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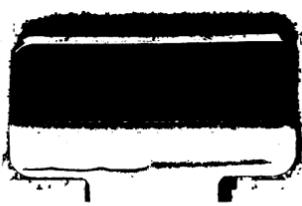
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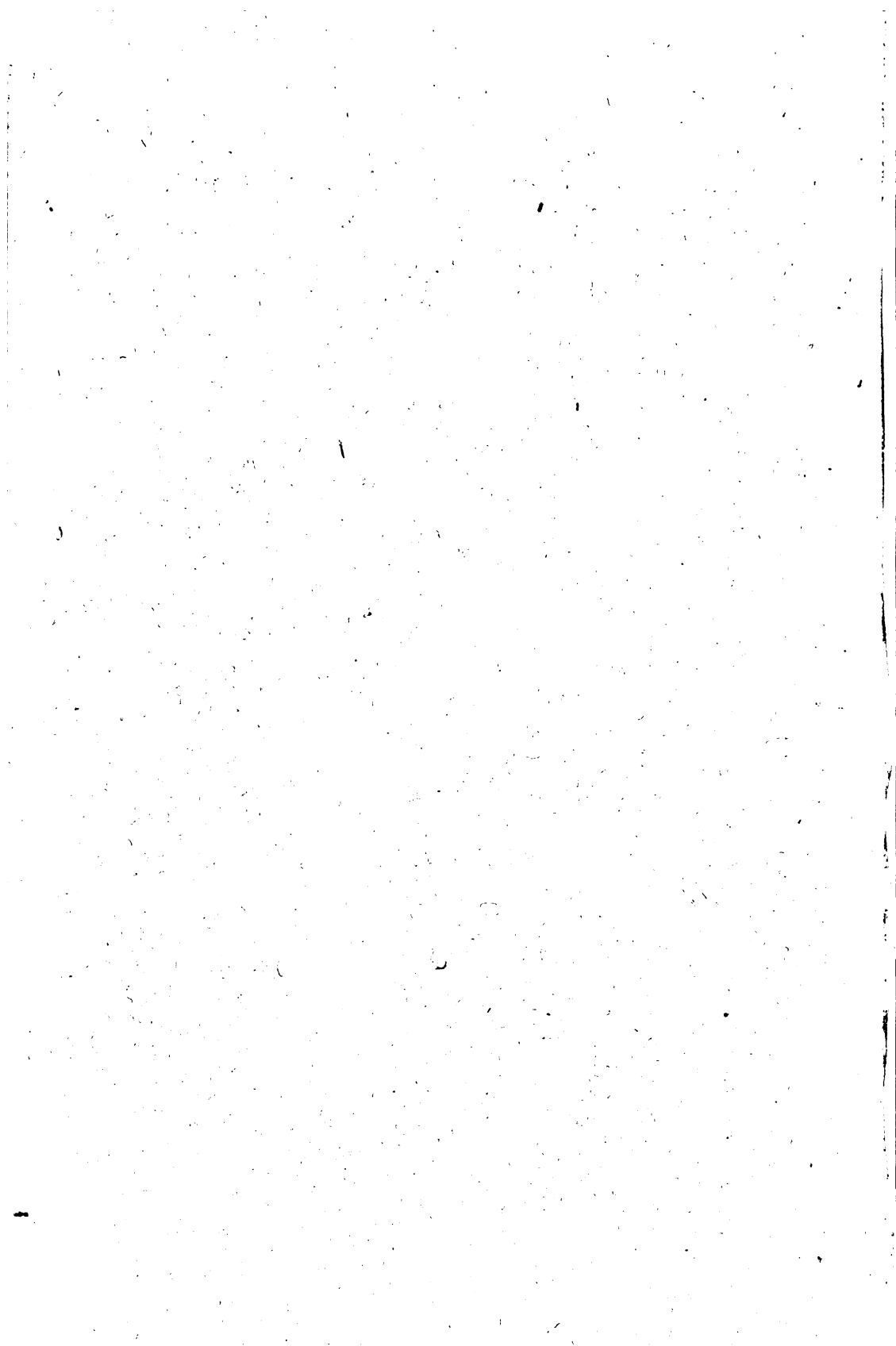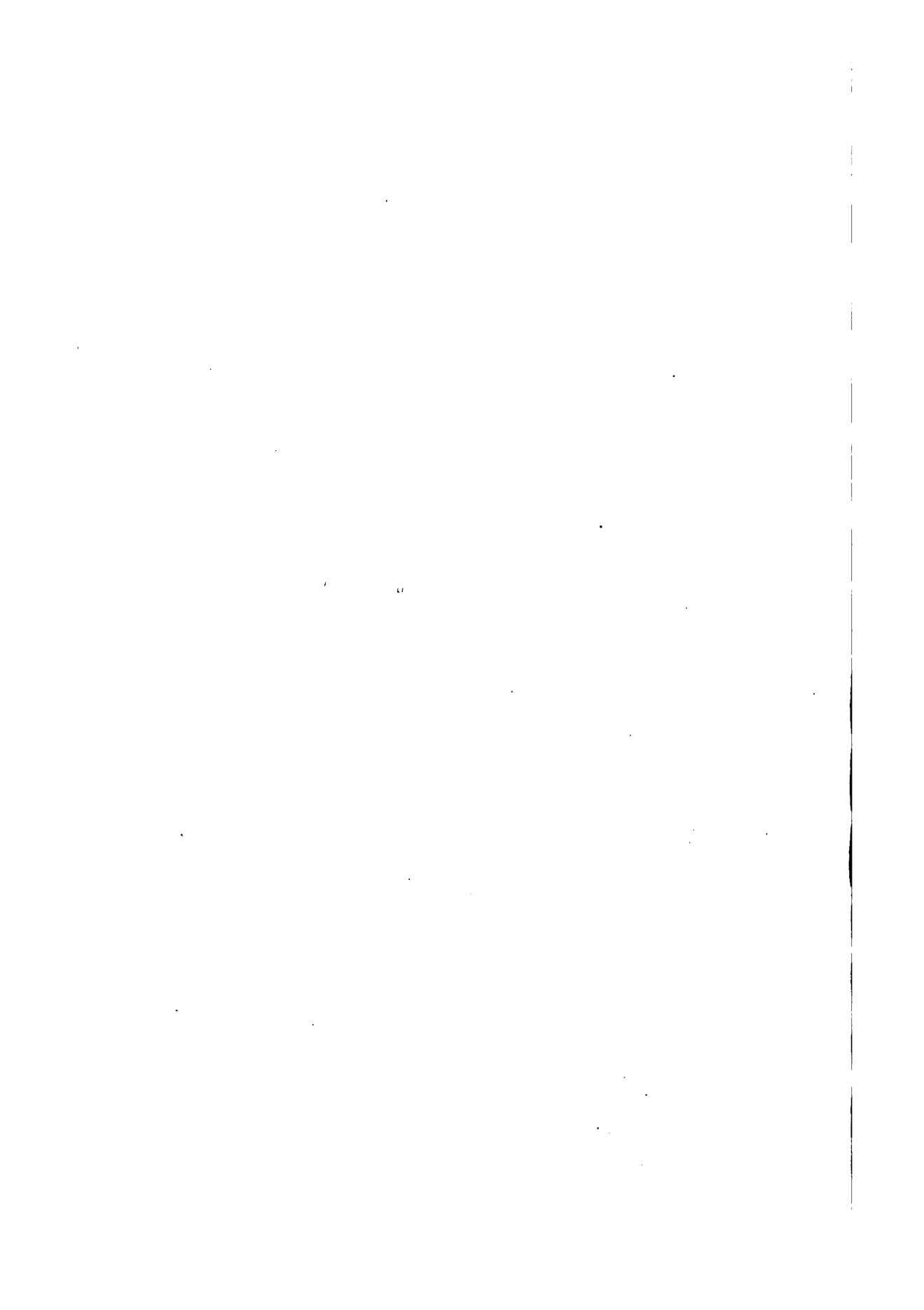
THE ECONOMY FACTOR

IN

STEAM-POWER PLANTS

THE ECONOMY FACTOR

IN

STEAM-POWER PLANTS

BY

GEO. W. HAWKINS

PREFACE

THIS work was prepared principally for the use of engineering students, and for that class of engineers having to do with the economical design of steam-power plants. The problems involved are now recognized by the entire profession as among the most important to be met. From a commercial view-point, the possibility of a future scarcity of fuel and the certainty of increasing fuel costs lend additional weight to the subject.

So far as is known, there is not another single volume comprehensively treating this subject, which in view of its interest and importance is a matter of much wonder. The data used in compiling this treatise has been carefully selected from the results of actual experiments, all mere reports and unauthentic results being absolutely discarded. It is believed, therefore, to represent the only authentic collection of data upon the economic performance of the various power plant constituents.

The principal use to which the method for solving for power plant economy described hereinafter may be put, consists in predetermining the results to be derived from various classes of power plants under different conditions. These results may be used by engineering contractors in determining proper values for economy guarantees, by the consulting engineer in comparing the relative merits of various types of proposed installations, by the student in a study of power plant economics, and for various other purposes.

It had been the original intention to make the method applicable only to oil burning plants, but inasmuch as the same method might obviously be made to apply to any fuel whatsoever, it was decided to add such "conversion charts" as would be necessary to afford a ready means of converting oil to coal or wood results and vice versa. The method may be applied, therefore, to any fuel desired with equal facility.

That part of the work relating to boiler efficiencies refers primarily to oil burning practice. It was not deemed necessary to enter into a discussion of this question on the basis of solid fuels for the reason that ample data may be had upon this subject from a number of sources. Further, the average engineer has well in mind results to be obtained from the various grades of coal and different furnace arrangements.

Nothing new nor previously undiscovered is claimed for this treatise, but it is believed that a careful perusal of the methods outlined will result in a more orderly knowledge of the economic results of power-plant apparatus, and the manner in which they effect the combined economy of steam-power plants.

G. W. H.

INTRODUCTION

THE demand for higher economies in steam-power plant design is to-day engrossing the attention of our most capable and prominent engineers. In the majority of cases, it is the scientific determination of each and every element entering into a given installation, which governs the plant efficiency and hence the ultimate profits of the investor.

This "combined efficiency" of the modern steam-power plant is a factor involving so many variables, that the greatest caution must be observed in dealing with it. It can only be used as a basis of comparison when a statement completely defining the nature of all of its component parts accompanies it.

From the fundamental principle of conservation, we know that the energy put into a plant in any form or forms whatsoever, must be numerically equal to the output in energy units — not the net useful output, else the plant efficiency would be 100 % and this article would never have been written, but the gross output, which may be divided into several types of energy.

PLANT EFFICIENCY

If E represents the total units of energy supplied and E_1 the total units of output, then the above idea is represented by

$$E = E_1 \tag{1}$$

But the question here arises: — "Of what are E and E_1 composed?"

The only elements entering a steam-power plant are fuel, water, and air, each carrying a certain amount of potential energy.

As to the output, there is —

(1) Energy carried away by the dry chimney gases in the form of heat.

(2) Energy carried away by moisture in the chimney gases due to the steam used for atomizing the oil, moisture in the fuel, the humidity of the air, etc., in the form of heat.

(3) Energy lost by exhaust steam in the form of heat.

(4) Energy lost by friction and the mechanical imperfections of the working parts, in the forms of sound and heat.

(5) Energy lost by radiation and leakage, from boilers, engines, auxiliaries, and piping, in the form of heat.

(6) Energy carried off by the circulating or injection water in the form of heat.

(7) Energy sent from the switchboard in the form of electricity to be utilized for commercial purposes.

Let g = energy carried away by dry chimney gases;

m = energy carried away by moisture in chimney gases;

s = energy carried away by exhaust steam;

f = energy carried away by friction;

c = energy carried away by circulating or injection water;

r = energy carried away by radiation and leakage;

e = energy output from switchboard;

h = energy supplied by fuel;

a = energy supplied by air;

w = energy supplied by water.

Then from equation (1) we have —

$$h + a + w = g + m + s + f + c + r + e$$

from which

$$e = (h + a + w) - (g + m + s + f + c + r)$$

or efficiency equals

$$\frac{(h+a+w) - (g+m+s+f+c+r)}{h+a+w} \tag{2}$$

All of the factors in this equation vary greatly in magnitude with every change in the conditions appertaining thereto, and it shall be the purpose of this article to demonstrate a method of investigation of all of these elements for several sets of standard conditions and to indicate the method of solution for any special case.

At the outset, it might be stated that the important item of input is the fuel; the output item the electric energy. The design of a plant from an economic standpoint may be regarded as successfully accomplished when the former is of the smallest possible, and the latter of the greatest possible magnitude for the special conditions involved.

Fuel and Steam Consumption

In order to arrive at the probable fuel consumption of a given plant under a set of definite conditions, it is necessary to estimate the total quantity of steam required in order to fulfil those conditions, knowing which, the fuel necessary to produce such steam is readily calculated. Here the steam consumed by the auxiliaries comes into prominent play. The rough method is to estimate the hourly steam consumption of the

main engine, either from the builder's guarantees, actual tests, or a knowl-
edge of what might reasonably be expected from the type of unit under
consideration.   To this is added a certain percentage supposed to cover
whatever steam might be used by the auxiliaries, etc., which sum is taken
to represent the hourly requirement of steam for the entire plant.

But this method is very crude indeed, entirely neglecting, as it does,
such vital factors as the kind and character of auxiliaries, high or low
vacuum, the temperature of circulating or injection water, whether or not
a cooling tower is used, the head against which the circulating pump
must operate, etc.   This latter quantity varies, of course, greatly, and
as will be shown later, bears a very strong influence upon the total steam,
and hence the total fuel required.   It will thus be seen that the steam
consumption of the auxiliaries is proportional to that of the main engine
only when all of the particulars entering into the consideration are known.

AUXILIARIES

It is evident that the type of auxiliaries will vary considerably in
order to fulfil the various conditions attendant upon the installation of
power plants for different purposes and in different localities.   While
a certain set of auxiliaries may give excellent results in one plant, it would
be the poorest practice to install them in another locality where special
conditions are to be met.   Thus a thorough knowledge of the relative
economies of the different types of auxiliaries and their influence on the
total plant economy is of the utmost importance.

Whether, for instance, they should be steam driven, whether electri-
cally driven, or whether belt driven, are some of the problems to be con-
sidered, each having its own peculiar advantage.   In one case a closed
type of feed-water heater may be interposed in the exhaust line of the
main engine to advantage, while yet again the exhaust from the auxili-
aries only should be used as a medium for heating the feed-water.   An
open type heater can be advantageously employed for other conditions.
The influence of fuel economizers and superheaters upon the economy of
a power plant are also of great importance, especially so in view of the
modern trend towards high steam pressure and superheat.

All of these considerations will be dealt with in their proper turn.
In the discussion to follow, the economic results to be obtained from each
of the units entering into a complete plant will be reviewed, leaving the
various methods of combining these results for subsequent discussion.

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PART IV. COMPLETE PLANT ECONOMY
(VARIABLE LOAD)

CONCLUSION

PART I

INDIVIDUAL APPARATUS

BOILERS

To commence at the beginning, a few words will be said relative to the burning of California crude oil under the best types of boilers, and the efficiencies to be realized thereby.

Fuel

This problem is greatly simplified, owing to the almost constant calorific value of the fuel. From a number of authentic determinations, this value appears to vary between 18,500 and 18,850 B. T .U. per pound. Some five years ago at the time oil burning first came into general use on the coast, the Babcock & Wilcox Company delegated their chief engineer to conduct a series of boiler trials in and about San Francisco with the idea of determining the best method of burning California crude oil under the boilers of that company. During this series of experiments, which lasted about eighteen months, the following average heat values were obtained: —

$$
\begin{array}{r}
18,887 \\
18,897 \\
18,846 \\
18,870 \\
18,937 \\
18,832 \\
\hline
\end{array}
$$

Average 18,878

The remarkable regularity of these results is apparent, and the slight differences are undoubtedly due to errors of determination as much as to actual differences in heat values. These tests were made upon Monte Cristo oil from Kern River Oil Fields, being of about 15° Baumé gravity and 205° flash test.

1

Later determinations made upon a mixture of oil from several wells of the same district developed the following values: —

$$
\begin{array}{r}
18,819 \\
18,845 \\
18,947 \\
18,841 \\
18,743 \\
18,732 \\
18,905 \\
18,880 \\
18,889 \\
\hline
\end{array}
$$

Average 18,845

These figures were taken during a period of from six to eight months and vary but slightly from the Monte Cristo oil. The oil in this case was of about 17° Baumé gravity with a flash test of 150°.

In addition to the Kern River Fields, the most important are of the Coalinga and Santa Maria Districts. The oil from the former varies in point of gravity and heat value to a greater extent than that from any other wells. A test upon oil of 21° gravity and 134° flash test showed a calorific value of 19,120 B. T. U. per pound. It will be noted that this oil was unusually light. It contained some 3.3% benzine, 40% kerosene, 20% lubricant, and 25% asphaltum.

In contrast with this analysis, the writer, while in the same field, saw two wells producing oil of 12° Baumé gravity and containing 56% asphaltum. The owners of these wells claimed this oil to be the heaviest yet discovered and these statements most certainly seem logical, for, when a large quantity of this fluid was lifted from the sump on a 12-inch plank, it must needs be scraped off with a shovel, resembling tar more nearly than oil. Its viscosity was very great.

But both of these examples are very extraordinary, and, as a rule, the oil is very uniform in character. Its chemical composition varies not more than 5% from the following: —

Carbon	86%
Hydrogen..	12%
Sulphur...	1%
Nitrogen, Oxygen, and other incombustible matter	1%
Total	100%

The Bakersfield or Kern River Oil Fields, which are one and the same thing, have practically reached the end of their usefulness and at the present rate of consumption cannot last but a few years longer.

The Coalinga Fields are comparatively new and apparently are much greater in extent than were the Bakersfield wells.

The Santa Maria Fields have been discovered but a short time, and, according to all reports, are the largest fields ever found.

All of the oil produced from these fields is supplied by the various oil companies to power users and others at from 50 c. to $3 per barrel of forty-two gallons, depending upon the location of the plant, transportation facilities, and the quantity required, etc.

It might be advisable to state at this time that the most usual method of expressing the economy of a power plant on the Pacific Coast is in terms of Kw. hours delivered to the switchboard per barrel of oil burned. It is a matter of fact that while a barrel of light oil necessarily weighs less than the same volume of the heavier product, the total heat value is very nearly the same in both cases. This is explained by the fact that the lighter oils have a proportionally greater calorific value per pound. From this it will be seen that the economy of a given plant in Kw. hours per barrel of oil is practically constant regardless of any slight changes in the gravity of the fuel consumed.

METHODS OF BURNING

The efficiency of the boiler is greatly influenced by the method of burning this fuel. The burner employed, the furnace construction, the atomizing agent, etc., are among the items to be considered in this connection. In the majority of plants, the oil is ejected into the furnace above the grate bars, which are covered with fire brick. The oil is atomized either inside or immediately outside the burner by means of steam or air, and is there ignited, the gases passing through the boiler in the usual manner.

THE BURNER

There are a number of kinds of burners upon the market, several of which give excellent satisfaction. These may be divided into several types as has been done by the Department of Steam Engineering of the United States Navy. These types depend upon the atomizing agent used and the method of its mixture with the oil. In regard to the former, there are three types: —

    (1) Burners using steam from main boilers;

    (2) Burners using compressed air under high pressure;

    (3) Burners using air under low pressure.

The first type is used almost universally throughout the coast, having been found simpler, more flexible, more convenient, and more economical, when properly applied.

Figure 1 shows diagrammatically the two leading types of steam operated burners, and is self-explanatory.   These two types are distinguished only by the method of applying the atomizing agent, the steam and oil being brought into contact outside of the burner proper in one case, and inside in the other.   The oil is usually preheated to a temperature of from 120 to 160° Fahr., by means of the exhaust steam from the oil pumps, and is supplied to the burners at a pressure of about 40 pounds by gage, but due to the throttling effect of the oil valve at the burner, the actual pressure at the burner tip is generally much less.
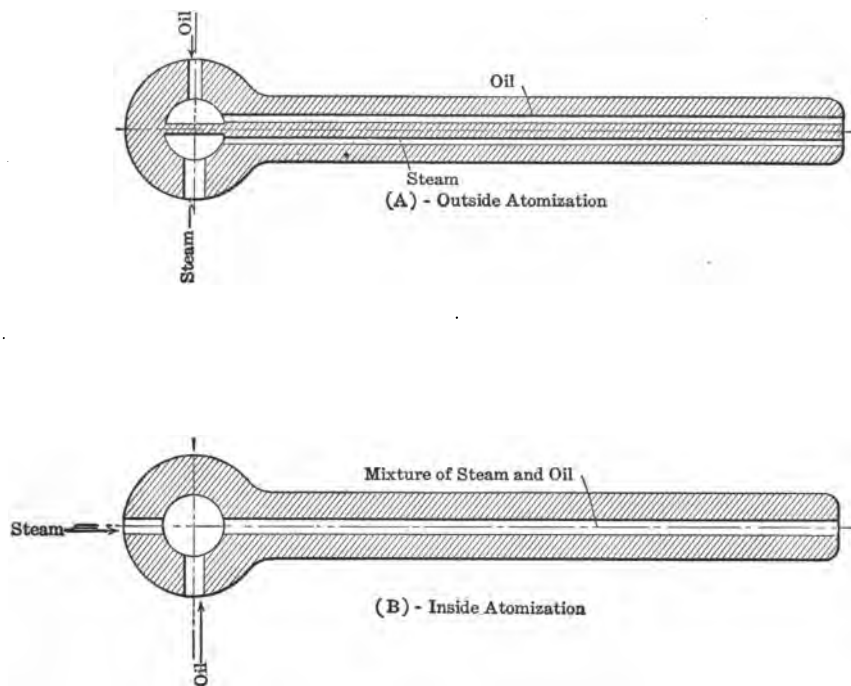
Fig. 1. — Showing Diagrammatic Arrangement of Fuel Oil Burners of the Inside and Outside Atomizing Types

There are many types of burners in use on the coast, but those above described are probably better known and are productive of better results with less attendance than any other forms on the market.

Furnace Construction

The furnace construction necessarily depends to a great extent upon the kind of boiler employed.   Externally fired tubular boilers are not adapted to oil fuel and are accordingly very inefficient when compared to the best forms of water-tube boilers.   Efficiencies have been obtained

as high as 73%, however, on large units under very careful manipulation. The highest type of internally fired tubular boiler — the Scotch Marine — while giving very fair results, falls below the water-tube boiler in point of efficiency as is conclusively shown by a large number of experiments during the past few years. They are ill designed for oil burning, partly on account of the limited furnace volume available, and partly because of the cooling effect of the comparatively cold water surrounding the furnace. Efficiencies have been obtained with the Scotch Marine boiler under favorable conditions as high as 78%.

What follows, will be construed as applying entirely to the best forms of water-tube boilers.

FURNACE VOLUME

First and foremost there must be provided a furnace of ample volume in order to obtain the proper diffusion of the gases and to insure complete combustion. This is of paramount importance. A large percentage of the failures in burning fuel oil are due to the fact that the ordinary furnaces designed for coal burning have been used without any modification. It is an admitted principle that the question of furnace volume is of great importance for any kind of fuel, but when oil is burned, its importance is increased tenfold. The furnace having sufficient volume to properly burn either anthracite or bituminous coal, is entirely inadequate for the requirements of oil burning.

AIR DUCTS

The admission to the furnace of air for combustion is another feature deserving of the most careful consideration. The most usual method is to admit the air through the ash-pit doors, and thence through an opening or number of openings in the fire-brick covering over the grate bars to the furnace, regulating its volume by the manipulation of the damper. The sizes and location of these openings regulate to a very large degree the shape of the flame, its distribution over the heating surface of the boiler, the rate of diffusion of the gases, the temperature of the flue gases leaving the boiler, and finally its operating economy. The proper and rapid diffusion of the gases means perfect combustion, while their distribution over the entire heating surface, together with a minimum excess of air supply over the theoretical amount required, make for low stack temperatures and high efficiencies. Carefully made tests have demonstrated that with well proportioned and located air openings, there is required not more than 10% excess air supply to furnish the necessary oxygen for the complete combustion of the fuel.

It should be remembered that this air must be admitted to the furnace in more or less large quantities, hence the difficulty of using all of the

air supplied, of allowing none to escape, and of obtaining complete and rapid diffusion of the gases before coming in contact with the heating surface of the boiler.

One ingenious method of insuring proper diffusion and which has been practiced with some success, is to construct the boiler setting with hollow side walls through which the air is passed before being admitted to the furnace. This method has two effects : —

(1) It considerably reduces the radiation loss;

(2) It supplies highly heated air for combustion.

The former loss alone represents from 2 to 6%, depending upon conditions, and is therefore worthy of consideration. Inasmuch as the greatest portion of this loss occurs from the side boiler walls adjacent to the furnace in ordinary practice, and this being precisely where the saving is effected, this method has much to commend it.

But as effecting the diffusion of the gases, we all know that the rate of this diffusion is a function of the temperature of the gases, varying in direct proportion with it. It thus appears reasonable to expect that complete combustion will be more easily and quickly accomplished by the use of such preheated air. That this theory is upheld in practice is a fact. Thus will be realized the importance attached to the scientific solution of the problem of admitting air to the furnace.

SHAPE OF FURNACE

As the process of combustion proceeds, the volume of the gases increases. For this reason, the furnace should be constructed in such a way that its sectional area becomes greater as it recedes from the point of entrance of the oil burner. This is impracticable in some forms of water-tube boilers, but it is to be noted that higher economical results have been obtained where this principle has been applied than in types of construction which render the adoption of this idea difficult or perhaps impossible.

INCANDESCENT BRICKWORK

Another feature of merit in furnace design is to so construct it as to present a large surface of brickwork which will become incandescent after the burners have been in operation a short time, thus tending to retain a hot fire-box and helping in this manner to produce perfect combustion. Any tendency towards cooling off of the furnace will inevitably result in the partial formation of carbon monoxide (CO) thus greatly reducing the boiler efficiency. This loss is very material, as can be seen by the following illustration: —

Chemists tell us that one pound of carbon completely burned to carbonic acid gas (CO_2) expels 14,600 B. T. U., but that when partially burned to carbon monoxide (CO) only 4450 B. T. U. are liberated.

The difference between these two values, viz.: — 10,150 B. T. U. represents the heat lost per pound of carbon, due to imperfect combustion, the carbon being only partially consumed. On a percentage basis, there is a net loss of $\frac{10,150}{14,600}$ or nearly 70% of the heat value of each pound of carbon as burned.

ATOMIZING AGENT

For atomizing the oil at the burners, either steam or air is employed, as stated above. In case steam is used and after the oil has been atomized, the necessary heat must be supplied to raise the temperature of the steam to that of the flue gases, which heat is simply carried up the stack and wasted. The theory that the hydrogen and oxygen in the steam are disassociated, the hydrogen being burned, and the oxygen helping to support combustion, thus adding to the heat value of the fuel, is erroneous. The steam used for atomizing simply passes up the stack in the form of superheated steam, and is thus a pure waste. This loss, however, when brought into the final column of the "heat balance," is so small as to be almost negligible.

In case air is employed, steam must be used to operate the compressor or blower, depending upon whether the high or low pressure system is utilized. While this method may possibly require less steam than where the latter is used direct as an atomizing agent, the first cost and the maintenance of an air system with its connections, its air pipes, its engine, and its many more complications, renders the final verdict much in favor of the direct steam system, which has absolutely nothing to keep in order save a few small pipes and valves. Furthermore, there is some doubt regarding the possibility of obtaining as high a degree of burner efficiency with air as with steam.

Superheated steam has frequently been tried for this purpose, having been passed through a system of piping in the furnace before being admitted to the burners. When very carefully guarded, a slightly higher economy has been obtained under test conditions than with saturated steam, but this gain has been so extremely limited, as to render the adoption of this method inadvisable. On the other hand, the action of the superheated steam appears to produce an unsteady flame — a rapid succession of small puffs rather than the steady, uniform condition which is desirable. Should one of these puffs extinguish the flame when the boilers were operating under a small fractional load, which is very possible, the oil would continue to flow into the furnace under the pressure of the oil pumping system, and coming in contact with the hot brick would form a large quantity of gas. This gas, when accumulated to the proper proportions, would explode with great violence and attending danger.

Several accidents have been caused by thus allowing the flame to be extinguished.

All things being considered, therefore, saturated steam at boiler pressure may be advocated as a highly satisfactory method of atomizing fuel oil.

FURNACE AND FLUE GAS TEMPERATURE

The higher boiler efficiencies to be attained by the use of fuel oil appear to be due not so much to the higher furnace temperatures as to the faculty of the gases to more quickly give up their heat on their passage through the boiler. A number of tests made with a Uchling Steinbart Pneumatic Pyrometer, showed a temperature of about 2700 to 2800° in the hottest part of the furnace of a 500 H. P. Babcock & Wilcox Boiler. Tests with a Chatalier pyrometer in a somewhat smaller boiler showed at one time a temperature of 3000°, which was the highest ever authentically recorded for a fuel oil furnace.

The temperature of the flue gases has been reduced when the air supply has been scientifically regulated to between 380 and 425° Fahr.

ULTIMATE EFFICIENCY

The above conditions have been set forth in detail as influencing each in a great measure, the ultimate boiler efficiency burning California crude oil. In a recent set of tests made at Los Angeles, the world's record of 83.3% [1] boiler efficiency was obtained, which is the more remarkable when it is considered that the maximum possible boiler efficiency theoretically is only slightly greater.

THEORETICAL BOILER PERFORMANCE

This will be more readily realized by the following calculation for the highest attainable results under ideal conditions. These figures will be based on the following assumptions:

Boiler pressure by gage..	150 lbs.
Flue gas temperature..	370°
Temperature of air in boiler room..............................	80°
Specific heat of flue gases.....................................	.24
Temperature of oil entering furnace............................	160°
Temperature of steam used for atomizing oil....................	212°
Pounds of steam required for atomizing per pound of oil.........	½

It is also assumed that the fuel will be completely burned in the furnace and that just sufficient air will be introduced to supply the theoretical

[1] Since writing the above, slightly better results have been obtained on large Babcock and Wilcox units.

amount of oxygen required — but no excess; further, that the fuel and air contain no moisture.

In regard to the above assumptions a few words of explanation may be in order. The assumed temperature of the flue gases is extremely low and cannot be attained in actual practice. The reason for this is obvious. Steam at 150 lbs. gage has a temperature of 366° Fahr. In order to retain the proper rapidity of water circulation in the boiler, it is necessary that the gases leave the boiler at a higher temperature than that of the steam. The assumed margin in this case is only 4° Fahr., which is very close.

The temperature of the oil entering the furnace is obtained by means of the exhaust steam from the oil pumps. While it is possible to obtain a higher temperature, such efforts are attended with more or less difficulty, not to say danger, due to the formation of gas, and the consequent intermittent action of the burners.

The steam used for atomization is greatly reduced in pressure and temperature before entering the furnace, due to radiation and leakage and the performance of mechanical work in the burner. It has been assumed, therefore, that this steam enters the furnace as dry saturated steam at a temperature of 212° Fahr.

One-fifth pound of steam per pound oil is a lower rate than has ever been obtained with any form of burner now extant. The best results to date show a steam consumption by the burners of about ⅓ lb. steam per pound oil.

While it is very possible to obtain perfect combustion in a well designed furnace, this result is always attended with the admission of an additional supply of air over and above that theoretically required. The combination, therefore, of perfect combustion with just sufficient air to theoretically burn the fuel is accordingly a purely ideal condition, not to be reproduced in practice.

The assumption relating to the moisture in the fuel and in the air is also unattainable, as a certain percentage is always present in each of these media.

Upon the basis of the above rigorous conditions there would inevitably occur the following losses:

(1) Heat carried away by the dry chimney gases.

(2) Heat carried away by the water formed by the combustion of the hydrogen in the fuel.

(3) Heat carried away by the steam used for the burners.

(4) Heat lost by radiation.

The heat carried away by the dry chimney gases per pound of fuel may be readily calculated when we know:

(1) The specific heat of the gases.

(2) Their temperature.

(3) Their weight.

The first two of these items have been already assumed and it remains only to ascertain the weight of these gases for every pound of fuel burned.

Under the remarks appertaining to California crude oil, its chemical composition was given as follows:

Carbon	86%
Hydrogen	12%
Sulphur	1%
Nitrogen, Oxygen, and other incombustible matter	1%
Total	100%

Carbon combines with oxygen when completely burned to form carbonic acid gas (CO_2) in the ratio of one part carbon to $2\frac{2}{3}$ parts of oxygen by weight. It therefore requires $2\frac{2}{3}$ lbs. of oxygen to completely burn one pound of carbon. The oxygen in the air, however, represents 23.2% of its total weight, from which it is at once evident that in order to supply $2\frac{2}{3}$ lbs. of oxygen, there will be required

$$\frac{2\frac{2}{3} \times 100}{23.2} = 11.5 \text{ pounds of air.}$$

Similarly we can figure that hydrogen combines with oxygen when completely burned to form water (H_2O) in the proportion by weight of one part of hydrogen to eight parts of oxygen, from which it is seen that for each pound of hydrogen consumed there will be required

$$\frac{8 \times 100}{23.2} = 34.48 \text{ pounds of air.}$$

Sulphur combining with oxygen in equal parts by weight to form sulphur dioxid (SO_2), requires for each pound burned in this manner

$$\frac{1 \times 100}{23.2} = 4.31 \text{ pounds of air.}$$

Obviously, the sum of the products of the percentage of each element in the fuel by the weight of air required for the combustion of each pound of that element will represent the total air required per pound of fuel. Numerically, this quantity will equal (.86 × 11.5) + (.12 × 34.48) + (.01 × 4.31), which, reduced, equals 14.0707 lbs. of air required per pound of oil.

To this must be added the weight of fuel consumed which is one pound, making the total weight of flue gases per pound of fuel 15.0707.

On the basis of these figures, the heat carried away by the flue gases per pound of fuel will equal the product of the specific heat of the gases

by their rise in temperature by their weight per pound of fuel. This is represented by

$$.24 (370-80) 15.07 = 1048.87 \text{ B.T.U.}$$

The heat lost by the formation of water from the hydrogen in the fuel may be conveniently estimated as follows:

As stated above, one part of hydrogen unites with eight parts of oxygen, forming nine parts of water by weight. One pound of oil contains .12 lbs. of hydrogen from which it follows that for each pound of fuel burned the flue gases will contain $.12 \times 9$ or 1.08 lbs. of water. The hydrogen forming this water will enter the furnace necessarily at the same temperature as the oil, and the water resulting from the combustion will absorb heat as follows:

(1) The heat due to the difference in temperature between the incoming oil and the boiling point of water at atmospheric pressure.

(2) The latent heat of evaporation.

(3) The heat due to superheating the steam thus formed to the temperature of the flue gases.

This quantity is represented by the following equation:

$$H = .12 \times 9 [(212 - 160) + 966 + s (370 - 212)]$$

in which

H = the total heat lost; and

s = the specific heat of the steam.

Ordinarily, the specific heat of superheated steam has been regarded as a constant (.48) for all temperatures, but recent experiments have confirmed the long-felt conviction among engineers that this assumption is in error. Taking the results of these experiments which have been conducted both in the United States and Europe, the instantaneous value for the specific heat of steam at any degree of superheat is given by

$$s = .00222 T - .377 \qquad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

whence the mean specific heat between any degree of heat and the saturation point equals

$$.00222 \frac{T + t}{2} - .377 \qquad\qquad (4)$$

in which

s = specific heat of steam at any temperature;

T = temperature of superheat;

t = temperature of saturation.

From this it will appear that the value of s increases with any increase in the degree of superheat and with the steam pressure, being much less than .48 for low values of T and much greater for higher values.

Applying this formula to the present conditions, the specific heat of the steam formed by the presence of the hydrogen in the fuel equals

$$.00222\frac{(370 + 212)}{2} - .377$$

which reduced gives .27. Substituting this value for s in the above equation representing the loss due to the formation of water and steam from the hydrogen in the fuel, it is found that this total loss which cannot be lessened in any way whatsoever save by changing the percentage of hydrogen present equals . . . . 1145.88 B. T. U.

The steam used for atomizing the oil will waste the following quantity of heat for each pound of oil atomized:

$$(370 - 212) \times .27$$

which equals . . . . 42.66 B. T. U.

It is possible that by means of hollow boiler walls, through which the air for combustion passes, boiler radiation may be reduced to about $1\frac{1}{2}\%$. If this loss were still further reduced — to say $\frac{1}{2}\%$ — there would then be radiated per pound of fuel about . . . . . . .93.00 B. T. U.

Summing up all of the above losses, the following tabulation results:

	B.T.U.
(1) Heat carried away by dry chimney gases	1048.87
(2) Heat lost by the formation of water from hydrogen in fuel .	1145.88
(3) Heat lost by superheating steam used for atomizing oil	42.66
(4) Radiation loss	93.00
Total...	2330.41

or say in round numbers . . . . 2330 B. T. U.

If the above conditions were obtained with oil having a calorific value of 18,500 B. T. U. per pound, the heat actually absorbed by the boiler would be 18,500 − 2330 or 16,170 B. T. U. per pound of oil, and the efficiency would be $\frac{16170}{18500}$ or 87.3%. If the oil used contain 18,850 B. T. U. per pound, the heat absorbed by the boiler would be 16,520 and the efficiency would be 87.7%.

The results of this calculation will show that when, under actual working conditions, where all the items of loss mentioned above are actually greater, and in addition other losses due to the presence of moisture in the air and fuel exist, an efficiency of 83.3% is attained, the working limit is closely approached.

Figure 2 represents the efficiencies to be obtained under test conditions for various sizes of the better class of water-tube boilers at different rates of evaporation. Full rated load (100%) corresponds to the evaporation of 3.45 lbs. of water per square foot of heating surface from and at 212° Fahr. Most of these figures have been obtained from the results of actual tests, but in some instances the curves have been produced for the very light

loads. It will be noted that contrary to coal practice, the most economical rate of evaporation is between 10 and 15% above rated load.
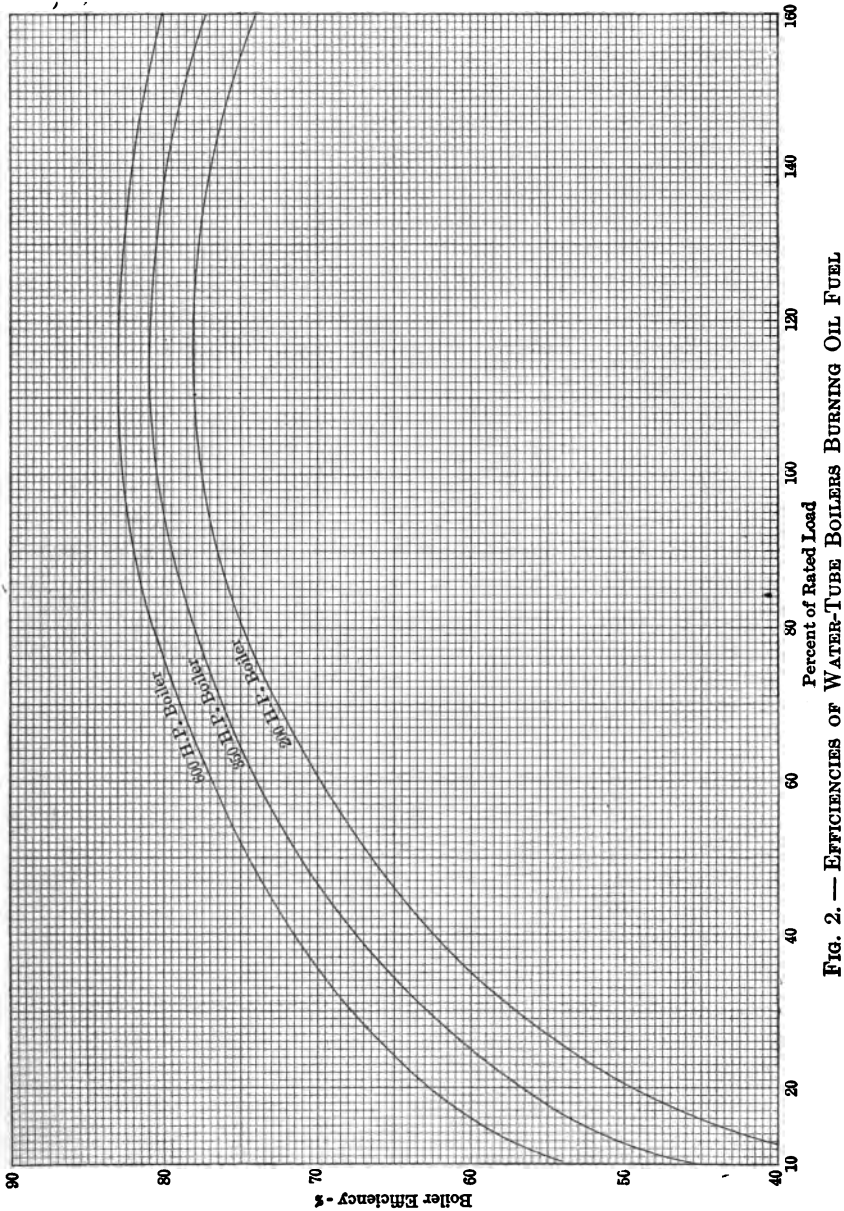
FIG. 2. — EFFICIENCIES OF WATER-TUBE BOILERS BURNING OIL FUEL

It has been deemed expedient to enter into the factors influencing the boiler efficiency, thus in detail, for the reason that oil burning and coal

burning practice are much at variance, and further on account of the fact that in the figures which are to follow, the boiler efficiencies employed might appear anything but conservative without the above explanations.

ENGINES

THE economy of the steam engine is the most important factor entering into the calculations for the resultant net economy of a complete steam-power plant. While it is out of the question to present an adequate tabular or graphical representation of the best results to be obtained from the various types of engines now predominant, due to the large number of variables which effect the final result, an attempt will be made to set forth in as compact a manner as possible sufficient data to enable one not in close touch with this subject to intelligently select at least approximately the proper value for a given set of conditions.

TYPES OF ENGINES CONSIDERED

Before arriving at numerical results, it will be necessary to review the types of engines to be considered and the elements which vary the working economy of the type selected. In this respect, attention will be confined to the following types of machines:

1. Automatic high-speed engines; 2. Medium speed four-valve engines; 3. Corliss engines; 4. Grid-iron-valve engines; 5. Steam turbines.

There are many other forms of engines, but the above are predominant in the lighting and power plants of this country. Poppet valve machines are not included as being of comparatively rare occurrence. The cheap forms of slide valve throttling engines with which the market is crowded are entirely ignored.

Each of these types has its particular field of utility for which it is better adapted than any of the other types. Occasionally, one comes into contact with an "enthusiast," prefixed by the word "high speed," "Corliss" or "turbine," as the case may be, but far more frequently the latter. This gentleman will magnificently claim absolute superiority for his special pet machine, regardless of conditions, and will confidently predict the rapid decline and final oblivion of all other types.

As a matter of fact, each has advantages and disadvantages peculiarly its own, and in order to intelligently arrive at the type best suited for any given requirements, a broad minded and unprejudiced view must be taken of all types of engines available, and due consideration given every condition to be encountered. This simply resolves itself into a case where commercial and engineering interests are identical as they always should be in any design. The writer has seen conditions such that the cheapest, most uneconomical machine could be installed to advantage, while under

different conditions unusual expense would be justified in obtaining the most economical, durable and reliable machine, embodying all of the refinements known to science for the purpose of realizing an apparently slight gain in economy.

But be it distinctly understood that a discussion of the relative merits of the various forms of engines is entirely outside the province of this article, except as relating to steam economy. There are, however, usually many other points to be considered in addition to that of steam consumption when finally selecting a unit for a definite service, and these are the considerations that are left untreated in this article.

Engine Economy

Having determined upon the proper type of engine required, the ultimate economy will vary, depending upon the following conditions:

1. Size of engine. 2. Temperature and quality of steam entering high pressure cylinder. 3. Temperature and quality of steam entering low pressure cylinder. 4. Point at which steam is cut off in high pressure cylinder. 5. Back pressure upon low pressure piston. 6. Ratio of cylinders. 7. Jacketed or unjacketed cylinders.

Each of the above conditions may materially affect the working economy of any type of engine under consideration. All of these conditions apply to each and every type of machine above noted, so it can readily be seen how difficult it would be to set forth the economy of each type of engine under the varying values for steam pressure, superheat, vacuum, etc., and also to take into consideration the effect of single cylinder and compound engines, both jacketed and unjacketed, etc. All we can do is to assume certain values as being in accordance with modern practice, and base our economy figures upon the results of available experiments under similar conditions.

Size of Engine

The larger the engine, in general, the more economical. This is due to many causes, principal among which is the fact that the radiation, leakage, condensation, etc., rapidly decrease in proportion as the size of the engine increases.

Temperature of Steam

The temperature of the steam entering the high pressure cylinder is one of the most important items in the determination of engine economy. This temperature is dependent upon the steam pressure carried and whether or not the steam is superheated. Superheat of a moderate temperature, say between 100 and 125° Fahr., has a very marked effect upon the resultant economy. This difference is much more marked

at fractional than at full loads, having a tendency to flatten the economy curve. It has been argued by some engineers that superheat increases the lubrication difficulties and that the increase in economy is not sufficient to offset the cost of the superheaters with their attendant fixed charges, repairs, etc. When it is considered, however, that practically every steam-power plant operates under a load factor considerably below 100%, and that the gain in steam consumption due to superheat is most marked at fractional loads, it will be found that superheated steam is almost invariably a paying investment. This necessarily depends, however, upon local conditions, price of fuel, etc., so that it is a consideration which must be figured separately for every set of conditions.

The item of repairs, etc., upon the superheaters, themselves, depends entirely upon the character of the superheater used. Separately fired superheaters are subject to considerable objection, arising from the fact that in addition to the radiation, stack losses, etc., in the boiler, all of these losses are duplicated in the superheating apparatus. On the other hand, in some forms of superheaters connected directly with the boiler proper, considerable difficulty has been experienced, owing to the design and location of the apparatus, the burning out of tubes, the inability to regulate the degree of superheat, etc. Inasmuch as superheated steam is practically a non-conductor of heat, it is readily seen that where this apparatus is exposed to the hottest portion of the flame, as, for instance, in or near the boiler furnace, the chances for difficulties are very much increased when compared with a superheater exposed only to a moderate degree of temperature. The large number of power plants in the United States in which superheaters are in use and the excellent results obtained in most cases would seem to be a leading argument in favor of superheaters.

We learn from a study of thermodynamics that the thermal efficiency of a perfect steam engine is represented by the expression

$$\frac{t_1 - t_2}{t_1}$$

in which t_1 = absolute temperature of steam at admission;
t_2 = absolute temperature of steam at exhaust.

While this relation is true for a perfect engine only, the same principle holds good in practice. From this it will be seen that efficiency varies directly as the difference in temperature of the steam at the points of admission and exhaust.

Summing up the results of experience in connection with this subject, it is safe to state that with the proper form or superheater, well designed, and with engines in which the valve gear is not affected by the high temperatures, marked economical advantages may be obtained by the moderate use of superheated steam.

REHEATING RECEIVER

In some plants it is a common practice to increase the temperature and quality of the steam entering the low pressure cylinder by interposing in the exhaust line between the two cylinders a superheating or reheating receiver. The gain in steam consumption to be realized by this means is entirely dependent upon the conditions under which the engine is designed to operate. For instance, in cases where there is a large ratio of cylinders, the reheating receiver is more marked in its effect than where the cylinders are nearer the same diameter. This refers to condensing engines particularly. There are also cases in which non-condensing engines may be fitted with reheating or superheating receivers to advantage.

CUT OFF

The point at which steam is cut off in the high pressure cylinder of a steam engine determines the horse-power developed and the working economy realized, other things being equal. Some discussion exists among engine builders as to the most economical point of cutting off steam, and it would appear that this varies with the different types and makes of valves and valve gears. In the figures to follow, it will be assumed that the full rated capacity of the engine is developed when steam is cut off at $\frac{1}{4}$ stroke, and this should be always borne in mind when using the accompanying graphs.

BACK PRESSURE

Inasmuch as the difference between the temperature of the steam entering the engine and the temperature of the steam leaving the engine is a measure of the thermal efficiency of the machine, it will be at once apparent that the greater the back pressure upon the engine, the more uneconomical the mechanism. In certain office building plants and plants used for industrial purposes in which the exhaust steam from the engine is utilized for heating the buildings or for drying, the back pressure of the engine is considerably in excess of that due to the atmosphere. In such cases, the economy of the machine is reduced below that of the ordinary non-condensing engine exhausting against atmospheric pressure. The other extreme is the condensing engine operating under high vacuum whereby the back pressure is reduced to a minimum. More will be said relative to condensing conditions under a separate heading.

RATIO OF CYLINDERS

The proper ratio of the high and low pressure cylinders of a steam engine is dependent entirely upon the conditions of pressure under which the engine operates. For high steam pressures, a high ratio should be

employed and vice versa. From this it will be seen that with a poorly designed combination of cylinders, the steam consumption of an otherwise excellent engine might be greatly reduced. Triple and other multiple expansion engines are not dealt with in this article as their use for stationary power and lighting service has been almost entirely discontinued, due to various reasons outside of economy of performance.

CYLINDER JACKETS

The effect of steam jackets upon the high pressure cylinders of compound engines is a subject regarding which there is still considerable discussion among engineers. Most engine builders, however, have found that when the steam admitted to the jackets is weighed in the balance against the economical advantage derived, that the result is not in favor of jacketed cylinders. This may have been brought about by the fact that it is much more difficult to secure good cylinder castings when the complication of jacket cores are added than for ordinary plain cylinders. It is a fact, however, that many engine builders who formerly supplied jackets invariably upon high pressure cylinders, have of late discontinued the practice entirely.

AUTOMATIC HIGH-SPEED ENGINES

Great strides have been made during the past few years in the economical development of the steam engine. Single cylinder piston valve high-speed non-condensing engines will now develop an indicated horse-power per hour on from 25 to 35 lbs. of dry saturated steam, depending upon the size of the engine and the steam pressure available. Compound engines of this type range between 20 and 25 lbs. non-condensing, and when condensing with 26-inch effective vacuum from 17 to 23 lbs. may be expected.

FOUR-VALVE ENGINES

Single cylinder non-condensing medium speed four-valve engines will range in economy between 21 and 26 lbs. of dry saturated steam, condensing between 17 and 22 per indicated horse-power per hour. Compound non-condensing engines of this type should be expected to operate upon from 16 lbs. to 21 lbs., and when condensing with 26-inch effective vacuum, from 13 to 17 lbs.

Corliss engine economy in pounds of steam per I. H. P. per hour very closely approximates the results obtained by the best forms of four-valve engines, and hence no additional figures need to be given under this head. It is a matter of fact, however, that owing to the higher speed and lighter working parts of the four-valve automatic engine, the friction of the machine itself is less than that of the heavy long stroke slow-speed Corliss machine, which means that in the case of the four-valve engine, the

mechanical efficiency is greater than that of the Corliss and hence the net economy per brake H. P. is somewhat better. For a guide in determining the relative economies upon the basis of brake H. P. output, it may be stated that in the best forms of self-oiling, four-valve engines, the mechanical efficiency should be between 93 and 94%, while in the most carefully designed and constructed Corliss engines, this value rarely exceeds 90%, and is usually less.

GRID-IRON VALVE ENGINES

Grid-iron valve engines of the type manufactured by McIntosh, Seymour & Co., are considered by many engineers to be the leaders in high-grade electrical work for large installations where economy is usually regarded as of first importance. Under condensing conditions and good steam pressure, the steam consumption per I. H. P. per hour varies between 14 lbs. for units of 500 Kw. capacity to about 12 lbs. in units of 2000 Kw. and above. Under superheated steam conditions, economy has been obtained upon the 5000 Kw. units of this type better than 11 lbs. of steam per I. H. P. The mechanical efficiency of these larger engines is over 96%. The horizontal vertical cross compound arrangement, in which the friction in the engine itself is reduced to a very small amount, owing to the arrangement of cylinders and the resulting uniformity in the crank effort, has shown a combined mechanical and electrical loss in direct connected units of as little as 6%, which means a net efficiency of both engine and generator together of 94%.

STEAM TURBINES

This brings us to the last and probably most discussed type of steam generator, namely, the steam turbine. The writer is very loathe to enter into the discussion of the comparative merits of the turbine and reciprocating engine. This subject has undoubtedly been the direct cause of more argument and criticism during the last few years than has fallen to the lot of any mechanical subject. It has only been during the last ten or twelve years that the turbine has occupied a prominent position in the engineering field. At first the advocates for this style of generator claimed economies 40% superior to those obtained by the best forms of reciprocating engines. Since that time, however, repeated experiments and developments gradually reduced this margin, until at the present writing there are comparatively few even amongst the most optimistic who claim to realize any marked economical advantages purely upon the grounds of steam consumption. Disinterested engineers must approach this subject with much caution and an utter lack of prejudice. Each necessarily speaks from his own experience. This much, however, may be regarded as absolute fact; — that in order to operate economically, the most im-

proved type of steam turbines, it is imperative that there be maintained:

    (1) High steam pressure.

    (2) Steam of a high degree of superheat.

    (3) And of the most importance, the highest possible vacuum.

Under these conditions excellent results have been obtained under a steady uniform load. It is, however, under a very variable load factor where extremely small fractional loads must be met that the steam turbine is distanced by its older competitor. This is diametrically opposite to the original claims made for the turbine, but if the reports of tests conducted by admittedly disinterested and competent engineers are to be regarded, the small amount of available data on station tests is consulted, and, above all, the actual guarantees made by the turbine manufacturers under competition for prospective plants are to be viewed as a criterion, claims for superior fractional load economics are entirely without basis.

ECONOMY CURVES

The presentation of complete data upon actual steam consumptions of the different types of engines for all possible conditions is a physical impossibility with the very limited number of authentic tests which have been made to date. It is extremely difficult to obtain reliable information upon this point. Each engine builder claims for his particular machine the highest economic results, so that the statements coming from this source must usually be discounted. Steam engine tests to be reliable must be made by disinterested parties, thoroughly experienced and capable in this particular line of research. Furthermore, those interested in economy almost invariably desire information in connection with the most economical types of machines so that there is a relative scarcity of data upon the most wasteful forms of engines — high-speed non-condensing piston-valve machines, for instance.

Figures 3 to 11, inclusive, set forth graphically steam consumptions for the different types of engines above listed. These figures represent approximately the best results to be obtained for the conditions enumerated. Some caution should be observed in using these curves, for the reasons above specified. These figures are based upon the following considerations:

For exactly similar conditions of operation, the steam consumption of any type of engine will increase as the size of the engine decreases. Taking this principle as a basis, there has been selected for each type of prime mover an engine of such size as will represent as nearly as possible the average size of engine used in modern power and lighting plants for that particular type. An economy curve has been plotted for each engine thus selected, showing its steam consumption at different loads under a

set of reasonably favorable conditions of steam pressure, vacuum, speed, etc.   The curves thus plotted have been taken as a basis for the representation of engine economies for machines of varying sizes, which latter

FIG. 3. — ENGINE ECONOMY — AUTOMATIC SINGLE CYLINDER PISTON VALVE NON-CONDENSING ENGINES. BASED ON DRY SATURATED STEAM, 125 LBS. I. S. P.
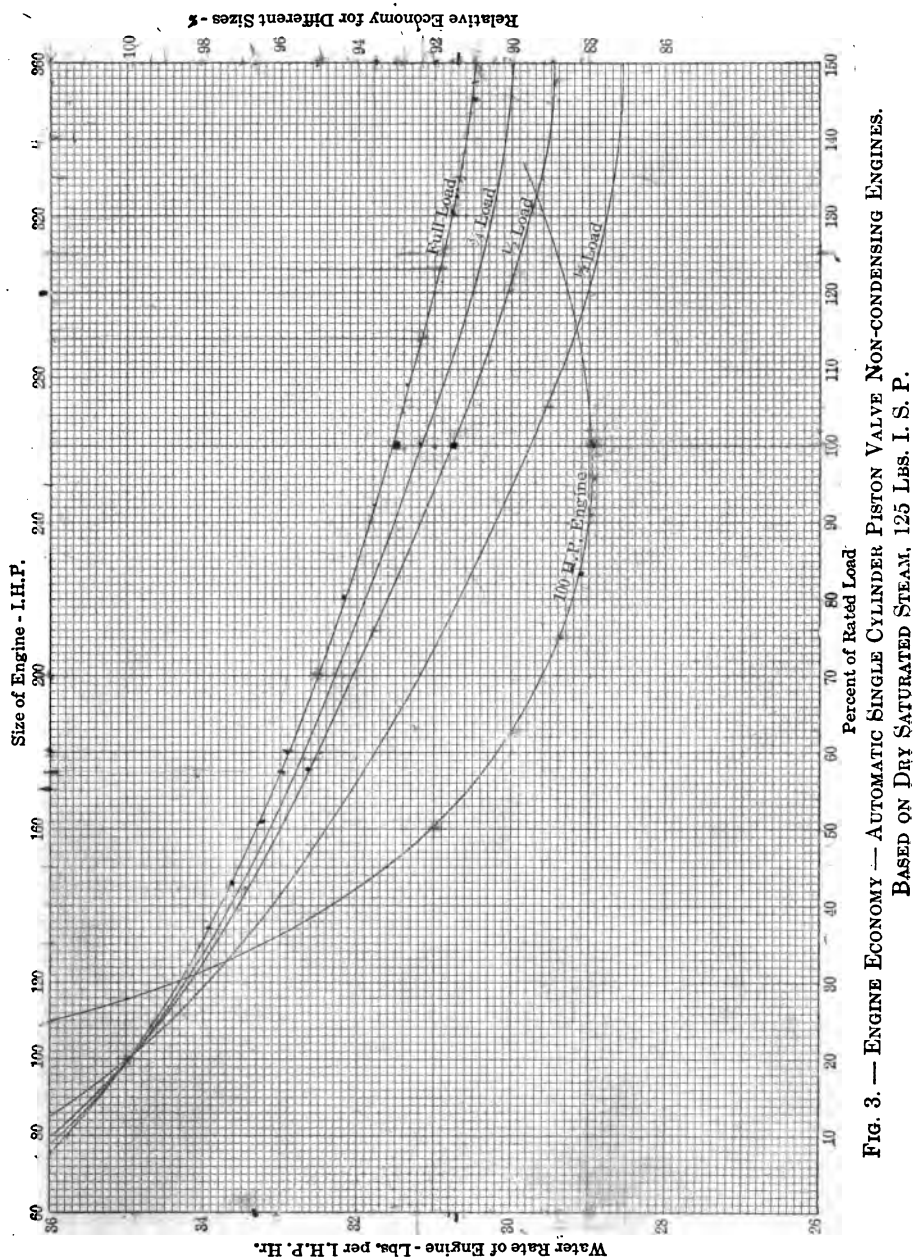
have been expressed in percentages of the economies of the selected and so-called "average" or datum engine. It might be well to state at this time that in nearly all cases, the accompanying figures have been taken
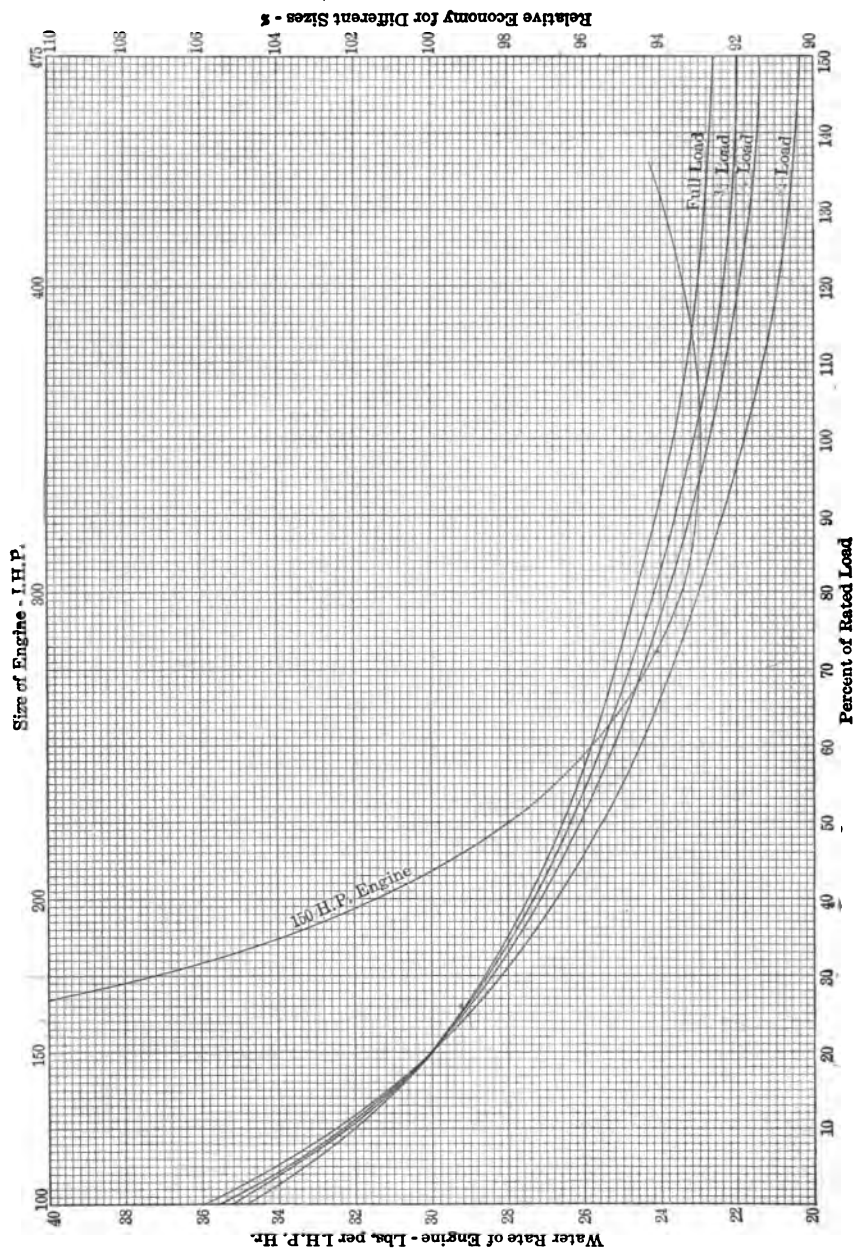
FIG. 4. — ENGINE ECONOMY — AUTOMATIC TANDEM COMPOUND PISTON VALVE NON-CONDENSING ENGINES. BASED ON DRY SATURATED STEAM, 140 LBS. I. S. P.

from actual tests under similar conditions.   In a few cases the results of
tests have been taken under slightly different conditions and the proper
corrections made to reduce the results to the required basis.   In a small
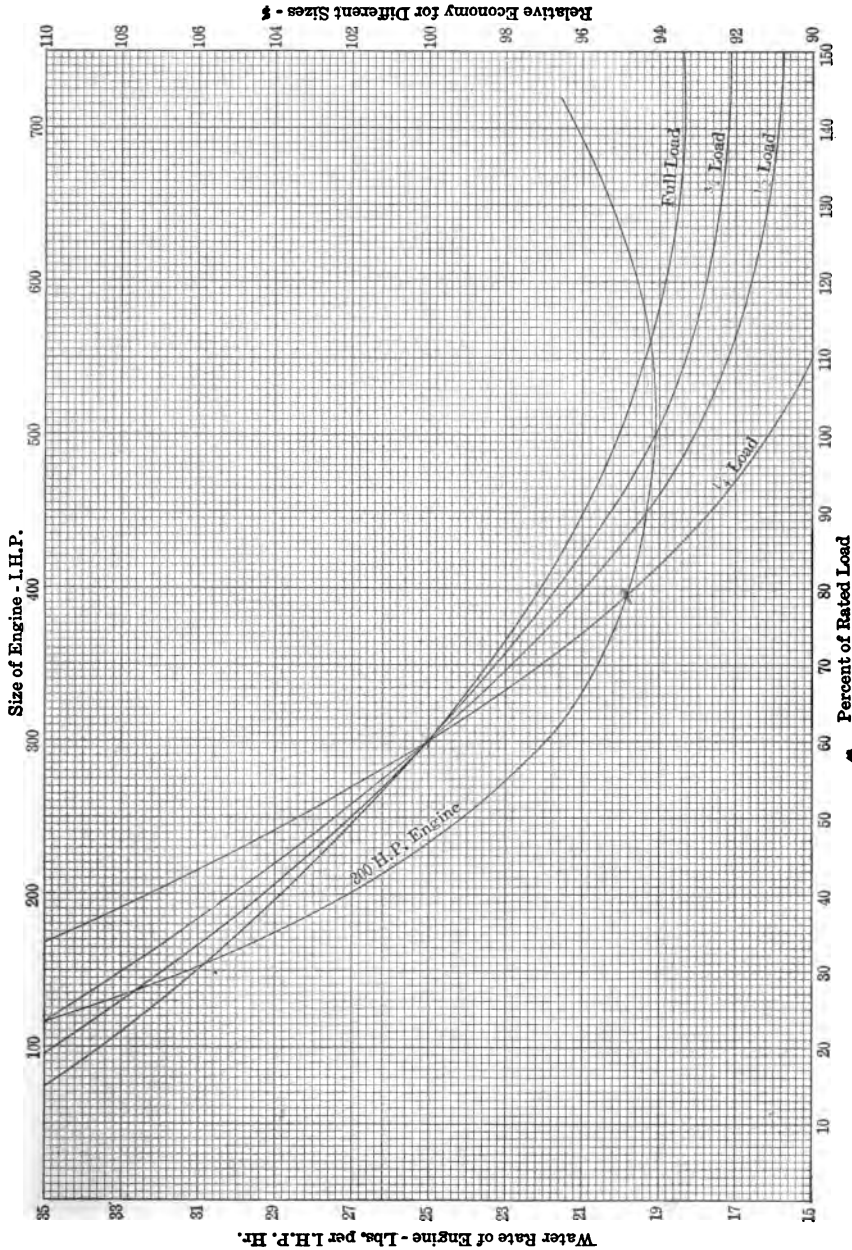
Fig. 5. — Engine Economy — Automatic Tandem Compound Piston Valve Condensing Engines.
Based on Dry Saturated Steam, 140 Lbs. I. S. P. and 26″ Effective Vacuum
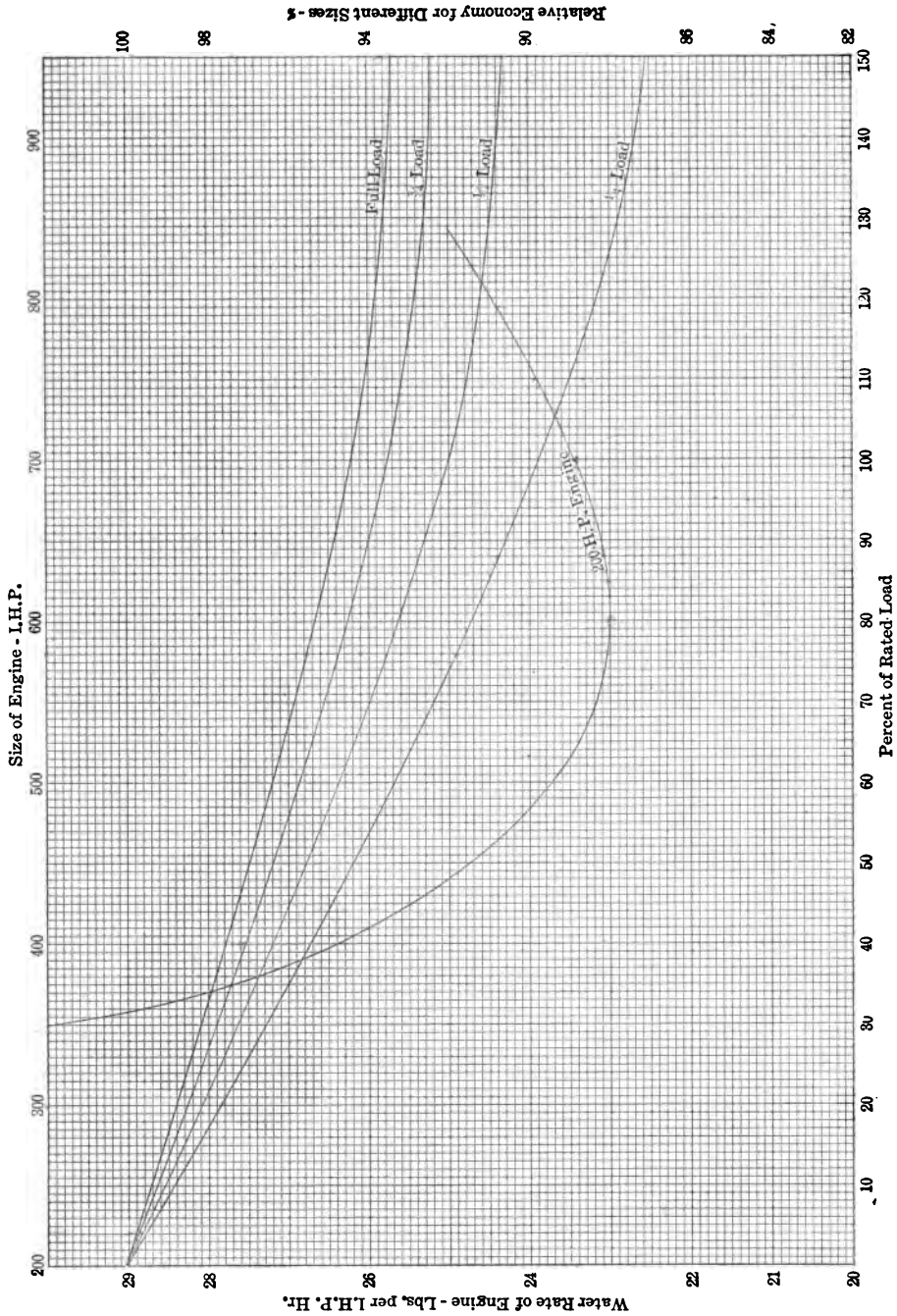
FIG. 6. — ENGINE ECONOMY — FOUR-VALVE OR CORLISS SINGLE CYLINDER NON-CONDENSING ENGINES, BASED ON DRY SATURATED STEAM, 130 LBS. I. S. P.

number of cases and in absence of authentic data near enough to the assumed conditions to be reliable, the curves have simply been produced in accordance with their general trend. This latter measure has been resorted to only as a final extremity, but in all cases where it has been
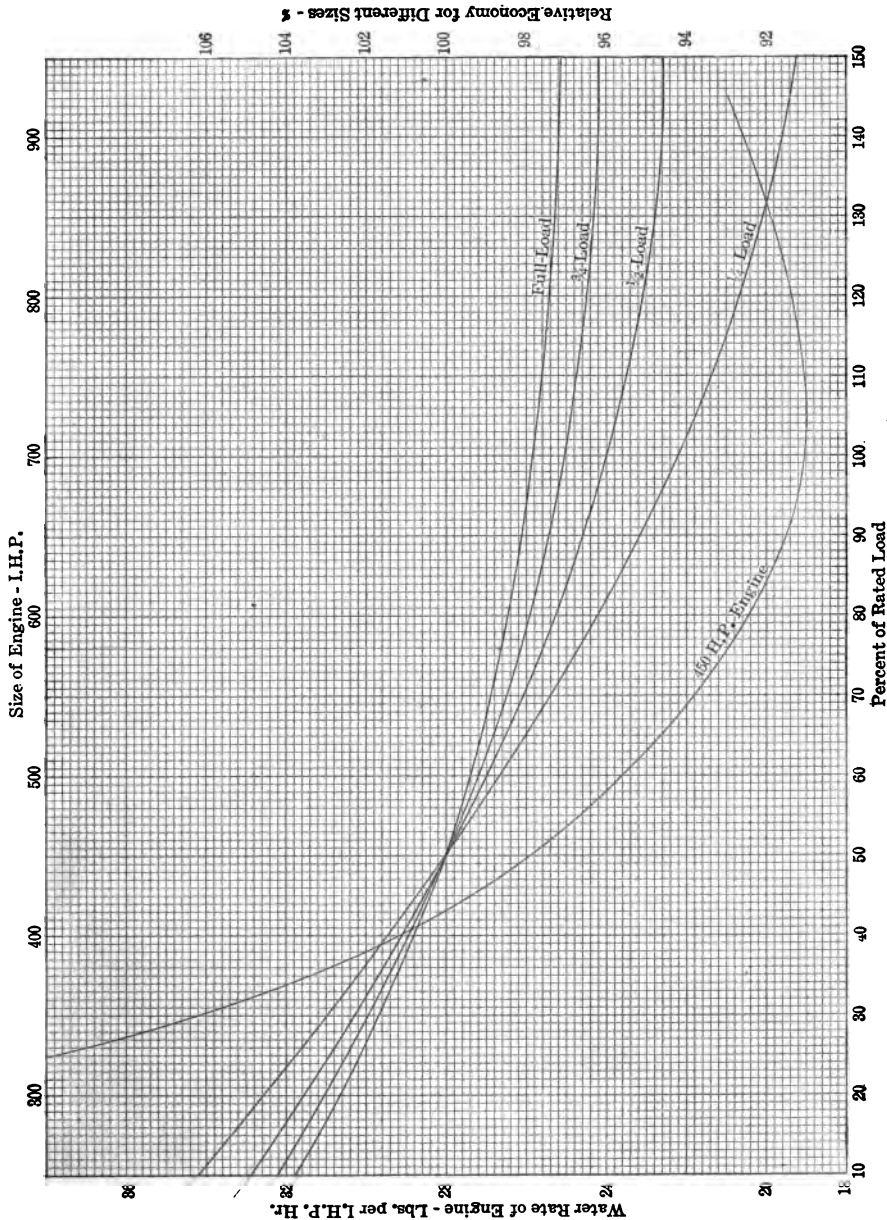
Fig. 7. — Engine Economy — Four-Valve or Corliss Compound Non-condensing Engines. Based on Dry Saturated Steam, 150 Lbs. I. S. P.

applied the results obtained check up closely with parallel data and are probably very close to the actual values.

Figure 3 represents the economical results to be obtained by automatic single cylinder piston valve non-condensing engines of the highest
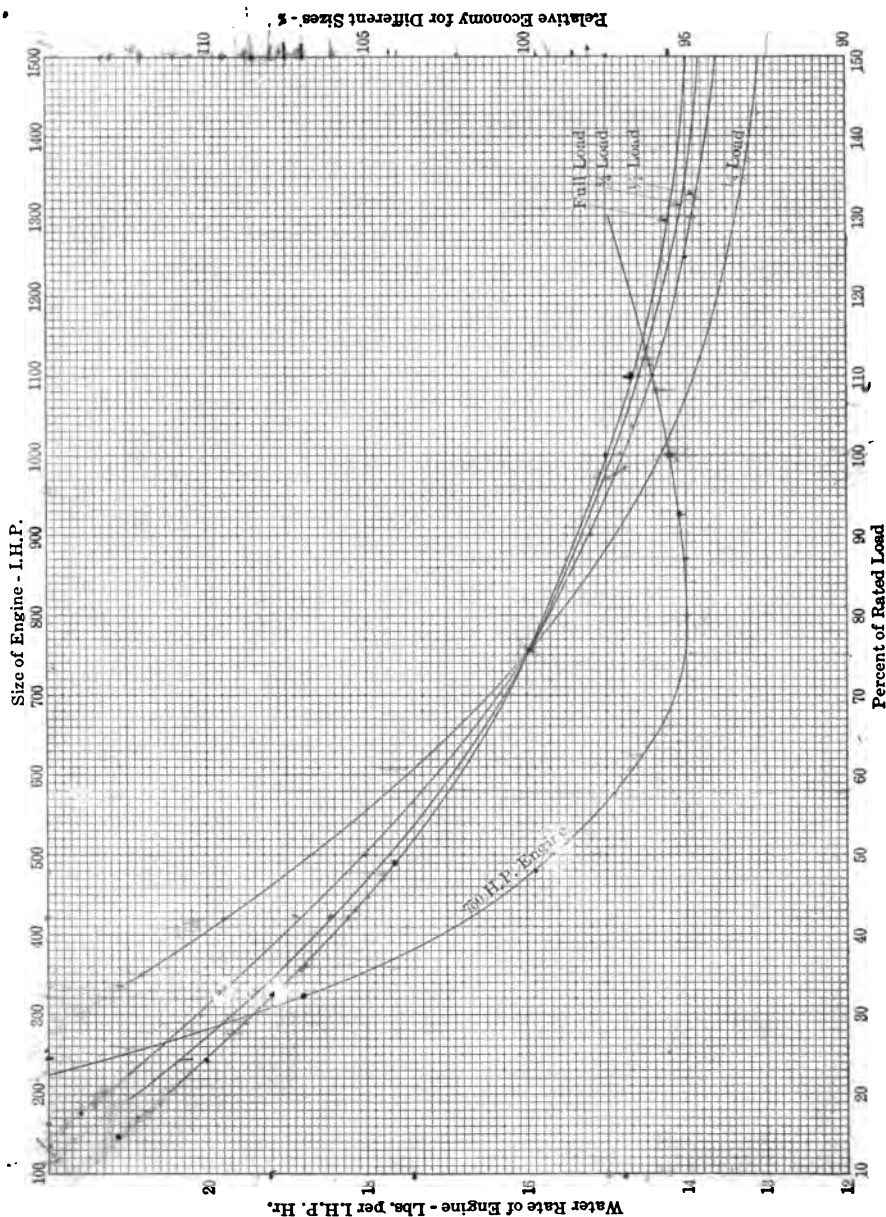
FIG. 8. — ENGINE ECONOMY — FOUR-VALVE OR CORLISS COMPOUND CONDENSING ENGINES. BASED ON DRY SATURATED STEAM, 160 LBS. I. S. P., 26" EFFECTIVE VACUUM
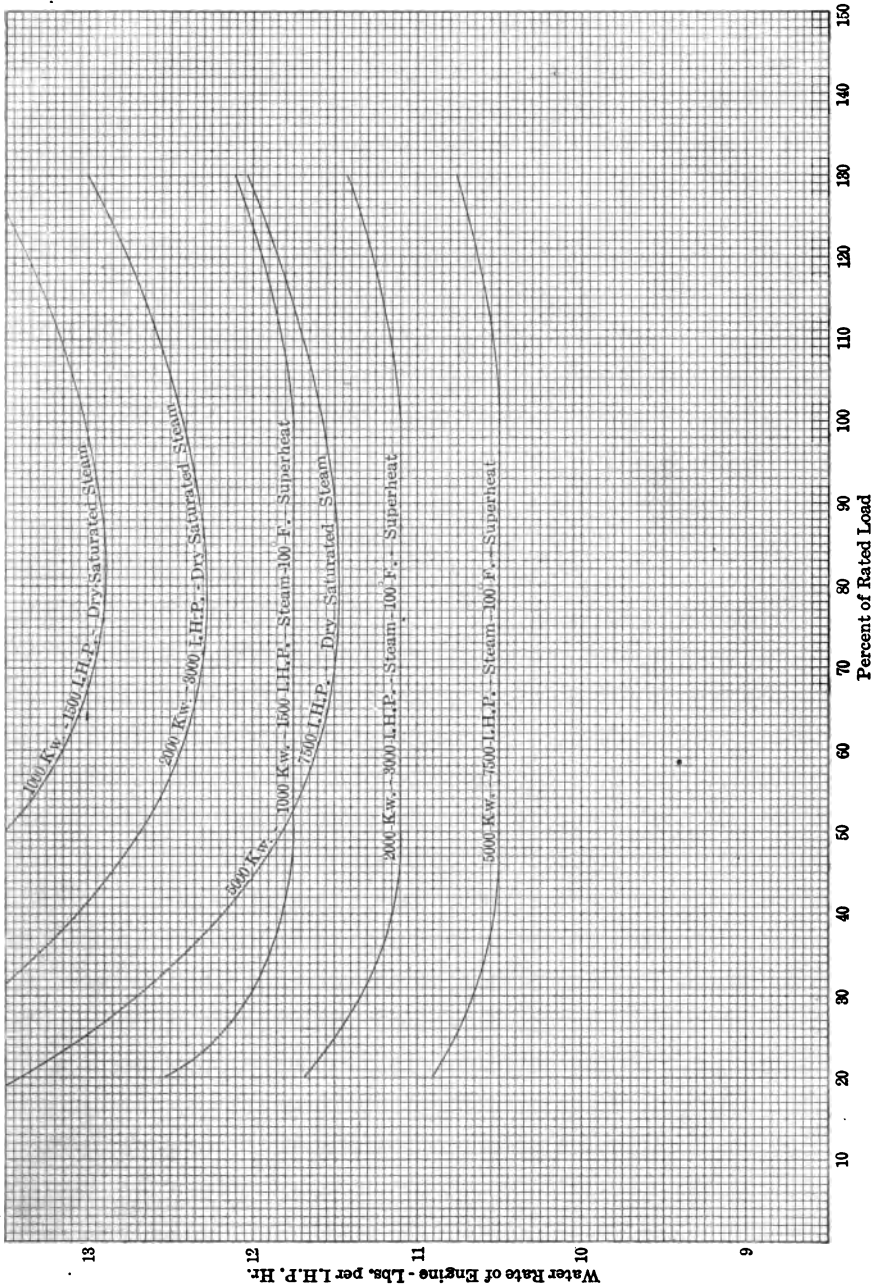
FIG. 9. — ENGINE ECONOMY — AUTOMATIC GRID-IRON VALVE CROSS COMPOUND CONDENSING ENGINES.
BASED ON 165 LBS. I. S. P., 26" EFFECTIVE VACUUM

class, using dry saturated steam of 125 lbs. initial steam pressure, and exhausting against atmospheric pressure.

Figure 4 sets forth the economy of compound piston valve non-condensing engines of the automatic type, using dry saturated steam of 140 lbs. initial steam pressure.

Figure 5 shows the steam consumption of compound piston valve condensing engines of the automatic type, using dry saturated steam of 150 lbs. initial steam pressure and operating with 26-inch effective vacuum in the low pressure cylinder referred to a 30-inch barometer. A cylinder ratio of 4 to 1 is also assumed.

In figure 6 is shown the steam consumed per indicated H. P. per hour for four-valve or Corliss single cylinder non-condensing engines based upon 125 lbs. I. S. P. and dry saturated steam.

The economy of four-valve or Corliss compound non-condensing engines, based upon dry saturated steam of 150 lbs. I. S. P., is shown upon Fig. 7.

Figure 8 shows the economy of four-valve or Corliss compound condensing engines operated with dry saturated steam of 160 lbs. I. S. P., and 26-inch effective vacuum in the low pressure cylinder, cylinder ratio 4 to 1.

With 165 lbs. I. S. P., and 26-inch effective vacuum, compound grid-iron valve engines equipped with reheating receivers may be expected to operate with economies set forth in Fig. 9. The three lower and more nearly horizontal curves represent the same conditions with 100° Fahr. of superheat added to the steam. Attention is called to the excellent results obtained with this type of engine using superheated steam, especially at fractional loads. All results are plotted in terms of pounds of steam used per indicated horse-power per hour.

Figure 10 shows approximately the best results reported for steam turbines. The four upper curves in this figure show the results of actual station tests in units of from 500 to 5000 Kw. in size. All of these figures have been corrected so as to be equivalent to a steam pressure of 180 lbs., a vacuum of 28″ with steam of 100° Fahr. superheat. They are thus put upon exactly the same basis. The uniform and characteristic nature of all of these curves is both apparent and interesting. They show clearly the fallacy of the widely circulated claims for superiority of the turbine at fractional loads. The lower curve in the same figure shown in dotted lines shows the best results reported for shop tests of steam turbines of various sizes under full load conditions. It will be noted that in no case do the station tests compare favorably with the shop tests made by the builders. The excellence of the economy figures at and over full rated load should however be recognized.

A few miscellaneous curves are set forth in Fig. 11. This figure

Note: Full Lines = Actual Station
Tests at Various Loads.
Dotted Line = Shop Tests
at Full Load.

500 Kw.

1000 Kw.

2000 Kw.

5000 Kw.

Size of Turbine - Kw.

Percent of Rated Load

Water Rate of Turbine - Lbs. per Kw. Hr.
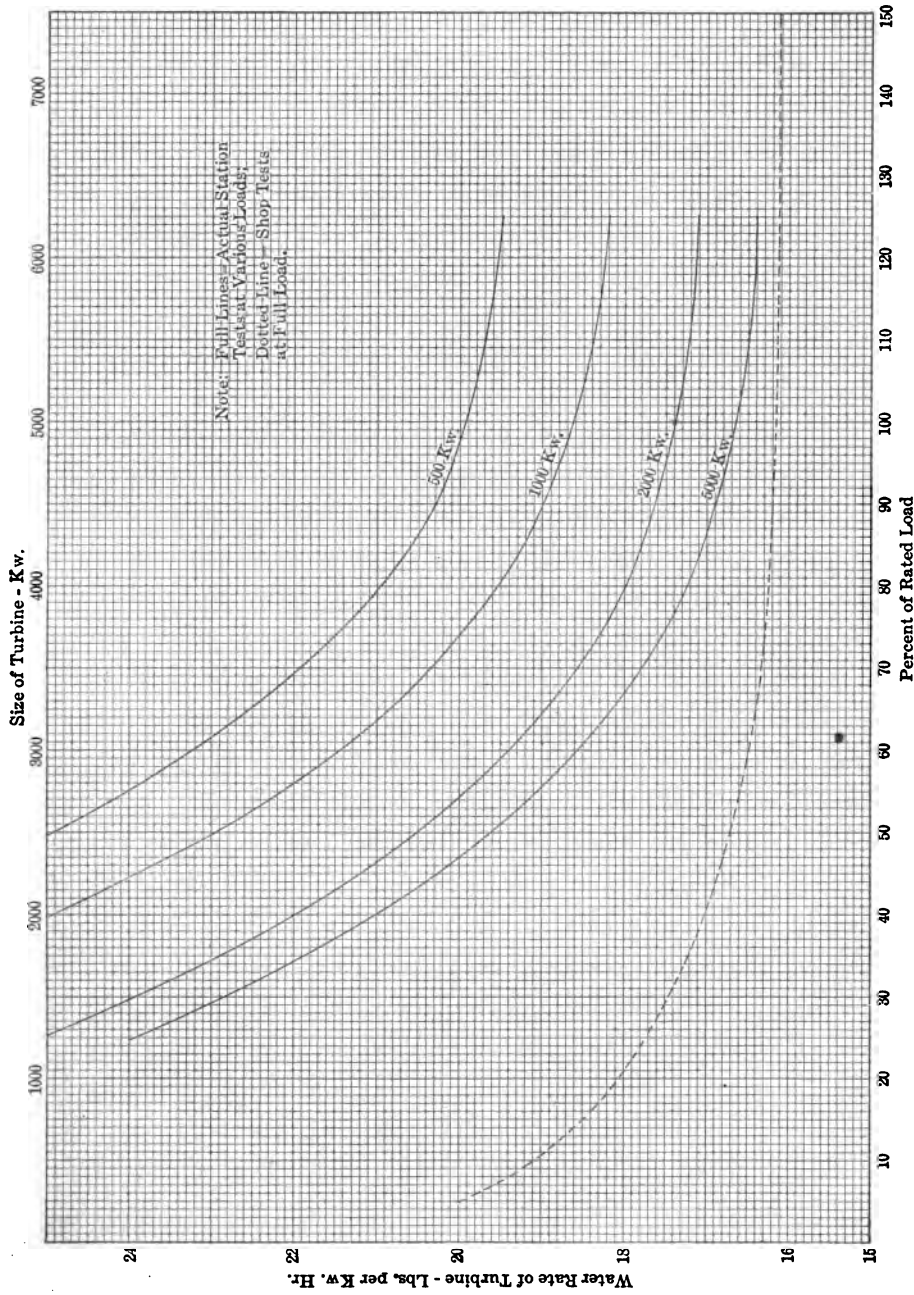
Fig. 10. — Engine Economy — Best Authentic Results Reported for Steam Turbines.
Based on Superheated Steam 100° F., 180 Lbs. Pressure, 28" Vacuum

brings forcibly to the front the actual comparison between steam engine and turbine economies. The turbine curve is the best authentic result thus far made public and shows the economy of a 5000 Kw. steam turbine with 180 lbs. steam pressure, 100° Fahr. superheat and 28-inch vacuum. The upper of the two engine curves shows the average of three station tests on grid-iron valve engines using about 155 lbs. steam pressure, 90° Fahr. superheat, and 26-inch effective vacuum. These three engines were installed in the same plant, each unit being only of 1600 Kw. capacity. The lower engine curve is representative of the steam consumption per Kw. hour for a 5000 Kw. grid-iron valve engine, based upon known values of the steam consumption per indicated horse-power per hour; known mechanical efficiencies of engine, and generator efficiencies as follows:

Load	Efficiency
$\frac{1}{4}$	89%
$\frac{1}{2}$	93%
$\frac{3}{4}$	95%
full	96$\frac{1}{2}$%
1$\frac{1}{4}$	96$\frac{1}{2}$%

The figures given also assume 170 lbs. initial steam pressure, 26-inch effective vacuum, and 100° Fahr. superheat.

To illustrate the manner in which the above curves are used, reference will be made to Fig. 8, which represents four-valve or Corliss compound condensing engines. It will be noticed that this figure is based upon a 750 I. H. P. unit, which is about the proper size for direct connecting to a 500 Kw. generator. Suppose it be required to find the economy of a four-valve compound condensing engine of 1250 I. H. P. at one-half, three-quarters, and full loads. From the figure will be obtained the following readings:

Per Cent. Rated Load	Steam Consumption 750 h. p. Engine	" Economy Coefficient " for 1250 h. p. Engine
$\frac{1}{2}$	15.75	95.2%
$\frac{3}{4}$	14.00	95.6%
full	14.20	95.8%

The product of the last two columns in the above tables will now represent the required economies of the 1250 H. P. engine which will leave the completed table as follows:

Per Cent rated Load	Steam Consumption 750 h. p Engine	" Economy Coefficient " for 1250 h. p. Engine	Steam Consumption
½	15.75	95.2%	14.99
¾	14.00	95.6%	13.38
full	14.20	95.8%	13.60

The above figures represent pounds of steam per indicated horse-power per hour, and are very nearly up to the limit of excellence of any Corliss engine economy.

Figure 12 is a graphical representation of the relation between the "economy coefficient;" the steam consumptions of the datum engine, and the required economy for a given engine. Having obtained from the proper curves the former values, the required result for any given engine may be read directly from this graph.

It should be remembered that full rated load is considered as being equivalent to 25% cut-off in the high-pressure cylinder in all curves for engine economy. Allowance should also be made in case actual conditions for which economies are required, vary from those set forth above.

ELECTRIC GENERATORS

THE efficiency of electrical apparatus plays an important part in the net economy of a lighting or power plant. As in the case of steam engines there are a large number of factors which effect the resultant efficiency of the machine. Different electrical characteristics exercise some influence. Very high voltages tend to insulation troubles and an attendant drop in efficiencies, etc. The speed of the machine is a considerable factor.

For accurate determinations, specific information should be obtained for each special case, but the accompanying Fig. 13 will suffice for approximate purposes. This represents roughly the electrical efficiencies realized for engine type alternators for all loads in sizes ranging between 50 and 5000 Kw. The dotted curve indicates full load efficiencies for machines of different sizes. All figures are based upon standard commercial machines operating at reasonably favorable speeds.

CONDENSING APPARATUS

THE idea of condensers as applied to steam engines is as old as the engine itself, and the economical advantages derived therefrom are too generally known and understood to require lengthy explanation here.

For the benefit of those not familiar with the subject, however, a brief investigation may be of value.

The horse-power of a steam engine is proportional to the product of the area of the piston, the piston speed and the mean steam pressure, from which it follows that by increasing any of these factors, either or
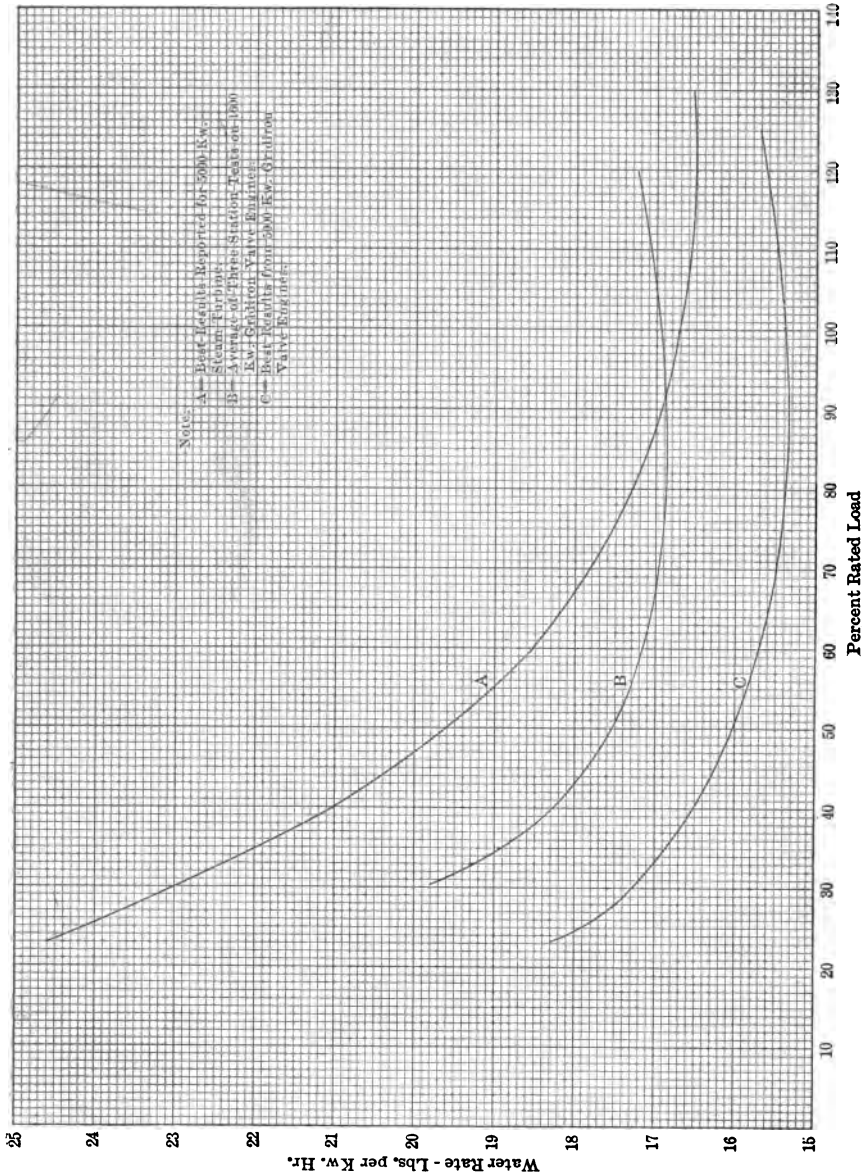
FIG. 11. — ENGINE ECONOMY — MISCELLANEOUS CURVES

both of the others may be decreased without changing the power output of the machine.   A non-condensing engine operates against a back pressure due to the atmosphere, which at sea level is equivalent to 14.7 lbs. per square inch.

This quantity, therefore, must always be deducted from the absolute mean effective pressure due to the steam, in order to obtain the net pres-
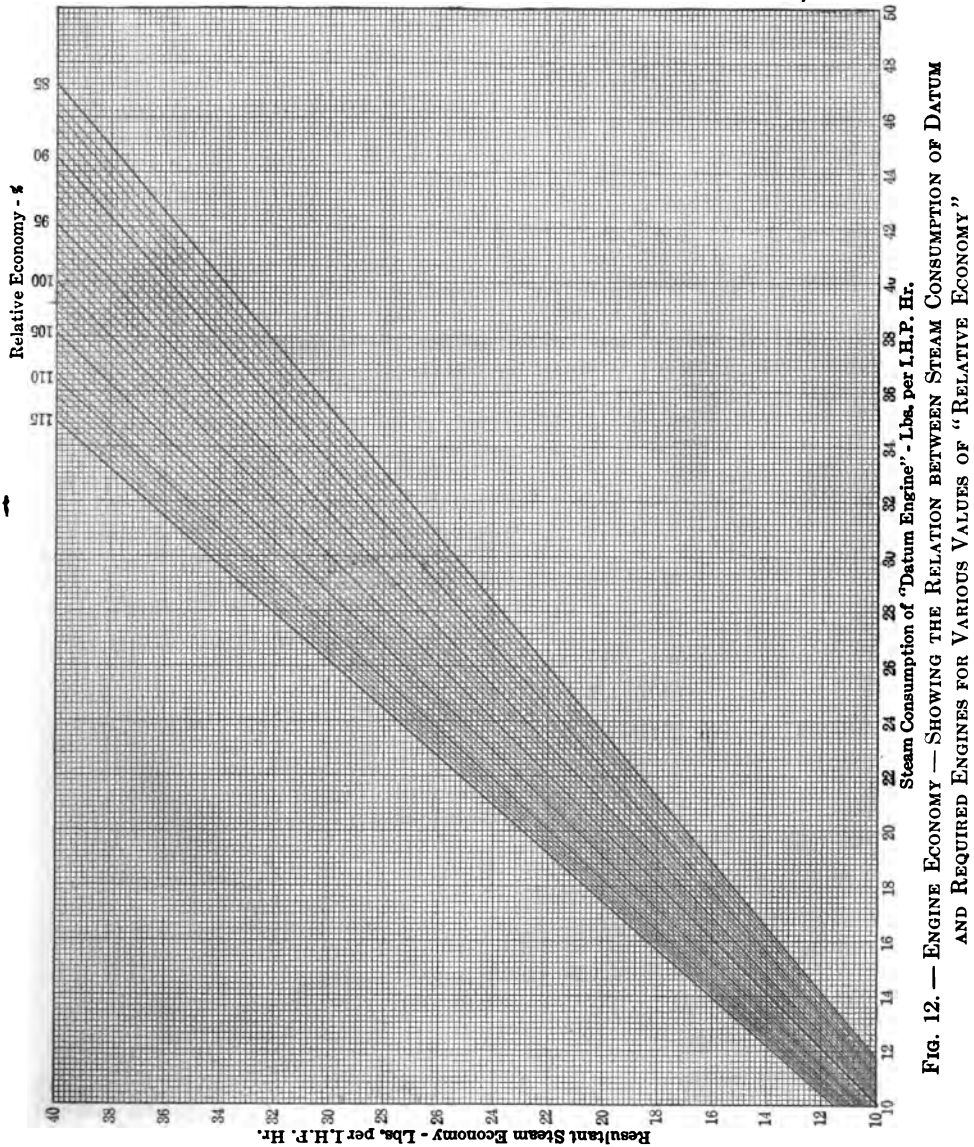
FIG. 12. — ENGINE ECONOMY — SHOWING THE RELATION BETWEEN STEAM CONSUMPTION OF DATUM AND REQUIRED ENGINES FOR VARIOUS VALUES OF "RELATIVE ECONOMY"

sure operating the engine.   It is therefore evident that in order to increase this net pressure, either the initial steam pressure must be increased or the pressure of the atmosphere must be removed from the piston.

The latter is accomplished by means of the condenser and the resultant gain in the steam economy of the engine will be clearly seen by a com-
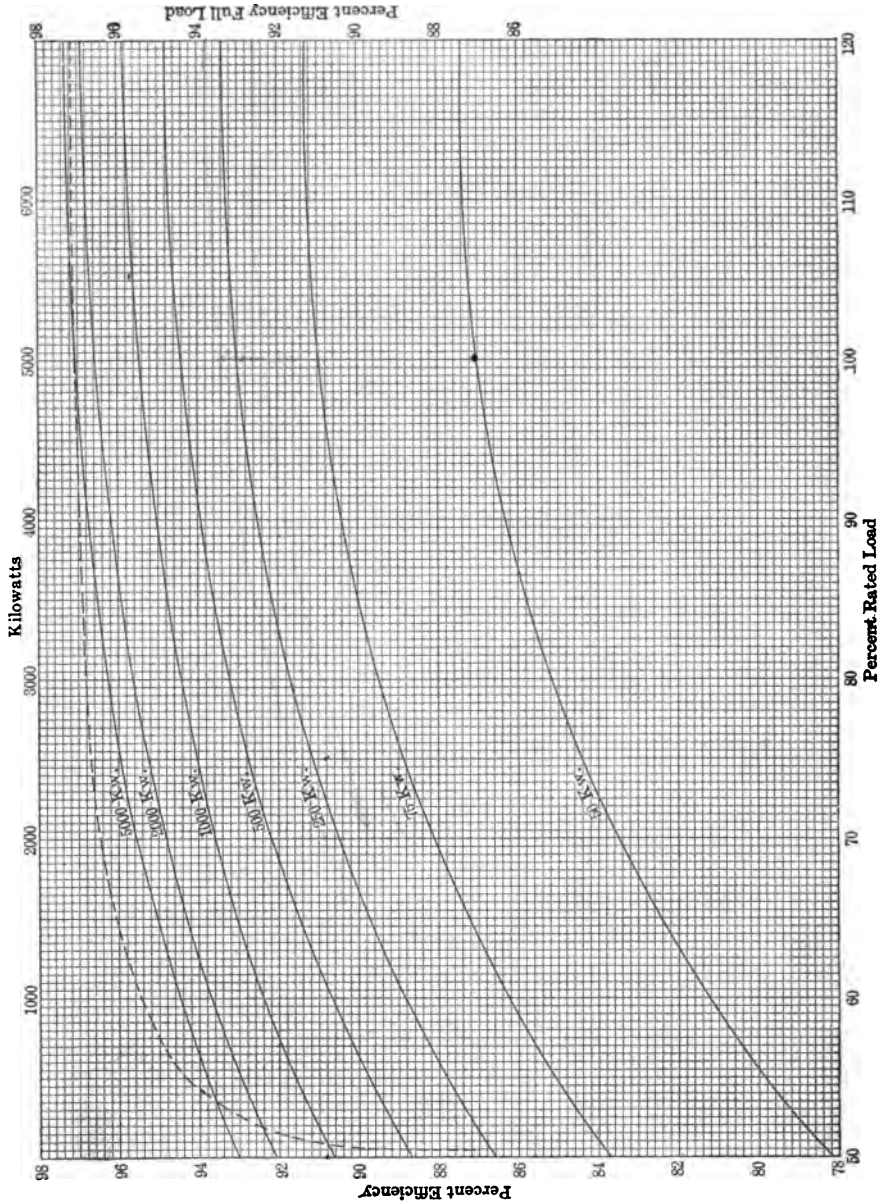
FIG. 13. — APPROXIMATE ELECTRICAL EFFICIENCIES FOR ENGINE TYPE ALTERNATORS

parison of Figs. 3 to 11. This results in a smaller boiler installation with proportionally smaller auxiliaries for a plant of given capacity.

To offset these advantages, a certain amount of steam is required to operate the auxiliaries incidental and necessary to the use of condensing apparatus.

In a jet condensing plant, this consists of an air pump, which must be of sufficient capacity to handle not only the entire condensation of the exhaust steam, with its accompanying air, but all of the injection water (which is from thirty to sixty times the weight of the condensed steam) as well as pumping the entire quantity against a certain head. Usually a separate injection pump is required for supplying the cooling water to the condenser in case no pressure from other sources is available. Each of these pumps requires a considerable quantity of steam.

Should a surface condenser be installed, there will be required, if the wet vacuum system is employed, an air pump for removing the condensed steam and its accompanying air, together with a circulating pump for supplying the cooling water; while if the wet and dry vacuum arrangement is preferred, a separate pump will be utilized for removing the air from the system. It will be impracticable to treat in detail the various modifications of the types of condensing apparatus above noted, due to space limitations.

The method of procedure for any special case will be similar to the methods hereinafter outlined, which will be confined to: (1), surface-condensing plants of the wet vacuum type; and (2), jet-condensing plants.

SURFACE–CONDENSING APPARATUS

In a surface-condensing plant of the wet vacuum type, the gain in steam consumption of the main engine is partly offset by the steam required to operate the air and circulating pump, so that it becomes necessary to know not only the increase in economy effected in the main engine, but also the steam consumed by the auxiliaries, necessitated by the addition of condensing apparatus; in other words, the steam consumption of the air and circulating pumps.

AIR–PUMP

By far, the greatest duty imposed upon the air-pump consists in removing the large amount of air present in the condenser. This quantity consists not only of the small percentage of air contained in solution in all water as was at one time supposed, but is mostly composed of the air leakage into the system through pores of castings, between joints, and into the low pressure cylinder of the engine through the stuffing box, etc. A conception of the amount of this leakage can be obtained when it is remembered that approximately 800 cu. ft. of air per hour will flow through

an orifice $\tfrac{1}{16}$ inch diameter under a pressure of 13 lbs., which is about equivalent to a vacuum of 26″. The importance of maintaining the tightest possible joints in the exhaust line, and of reducing all leakage to a minimum is therefore evident.

As the presence of this air in the condensing system excercises a prominent influence in the vacuum obtained, the exact amount of power required for the operation of the air pump is a question rather difficult of solution from a mathematical standpoint. Experience and observation show that this quantity should not exceed in well-designed plants .00024 H. P. in the air-pump cylinder for each pound of exhaust steam from the main engine per hour. The total power required by the air-pump represents a very small proportion of the entire plant output so that a slight error in this assumption has very little effect upon the net power-plant economy. From this figure, the total quantity of steam utilized by the air-pump is easily obtained for a given plant when the size and economy of the main engines, together with the efficiency and steam consumption of the air-pump, are known.

Direct acting single steam-driven air-pumps have an efficiency of not much over 60%, from which the I. H. P. in the steam cylinder of the air-pump per pound of exhaust steam equals

$$\frac{.00024}{.6} = .0004.$$

These pumps also have a steam consumption of from 100 to 150 lbs. of steam per I. H. P., depending upon the size of the pump, etc.

Vertical crank and fly-wheel suction valveless pumps of the Edwards pattern, while operating with but slightly greater efficiency, utilize only from 30 to 60 lbs of steam per I. H. P. per hour.

CIRCULATING PUMPS

Of far more importance than the steam requirements of the air-pump is that required by the circulating pump. While, as a general thing, high vacuums are advantageous, a point is finally reached where, owing to the large quantity of circulating water required and the corresponding amount of power necessary to pump it, the gain in steam consumption of the main engine is over-balanced by the steam required by the air and circulating pumps. The most economical vacuum for a given plant is reached at the point where the net saving due to condensing represents the largest possible interest upon the extra first cost of a condensing plant. This exact point depends upon the cost of condensing apparatus, the price of fuel, and the availability and temperature of circulating water. Despite the low temperature of the condensed steam incidental to the use of high vacuums, modern pratice has demonstrated that under

average conditions, an effective vacuum of from 25 to 26″ in the engine cylinder is most advantageous.

The quantity of circulating water required under these conditions varies between 35 or 40 to 50 or 55 lbs. per pound of exhaust steam, where the temperature does not exceed 60 or 70° Fahr., and increases to from 80 to 100 lbs. per pound of exhaust steam where the circulating water is high in temperature, as is characteristic of cooling tower practice.

Horizontal steam-driven direct-acting-single-circulating pumps have an efficiency of about 80% and require from 100 to 150 lbs. of steam per I. H. P. per hour.

Centrifugal circulating pumps should show from 40 to 65% efficiency. When steam driven, they are usually direct connected to single cylinder non-condensing steam engines, the latter having a mechanical efficiency of about 95% and a steam consumption of from 30 to 50 lbs. of steam per I. H. P. per hour. The net combined efficiencies of these direct connected units vary therefore between about 38 and 62%. The lower efficiencies refer to small pumps, operating against low heads.

ELECTRICALLY-DRIVEN PUMPS

Thus far, steam-driven pumps only have been considered, and, in general, these are preferable to electrically-driven auxiliaries. The use of such uneconomical types of steam-driven auxiliaries mentioned above would not be permissible under any circumstances were it not for the use to which the exhaust steam can be put in heating the feed-water, thus returning most of the heat contained therein, to the boiler. In selecting the auxiliaries for a steam plant, care should be taken to have them of such type and economies, when possible, as to make available only a sufficient quantity of steam to heat the feed-water to a temperature of about 212° Fahr. Any additional quantity of exhaust steam beyond this amount represents an unwarrantable waste, but if this quantity is not obtained, a considerable loss is sustained, due to the wear and tear on the boilers and sometimes to the resulting impurities in the feed-water.

In some cases, however, particularly where fuel economizers are used, it becomes desirable to employ power-driven auxiliaries, either belted or direct connected to motors. This is explained when it is remembered that the economizer efficiency is inversely proportional to the temperature of the entering water.

The efficiency of the air and circulating pumps is approximately the same, whether steam or motor driven. In the latter case, however, the net efficiency of the set may be obtained by combining the efficiency of the pump with that of the proper size of motor.

A much more extended comparison of the economic merits of steam-driven auxiliaries and motor-driven apparatus used in connection with

fuel economizers will be set forth in detail in the analyses to follow.   In the meantime, however, let it be remembered that the claim often made by feed-water heater dealers to the effect that practically all of the heat in the exhaust steam from the auxiliaries is returned to the boilers, the efficiency thus approaching 100% cannot be regarded as authentic and is not borne out in practice.

Radiation and leakage in small auxiliary steam pipes, condensation and leakage from pumps, leakage from poor joints, and, finally, the impossibility of proportioning the exhaust steam as nicely as to have just enough to heat the feed-water to its own temperature — no more or no less — all combine to render the actual working condition far from theoretical so that in reality the total thermal efficiency of steam-driven auxiliaries falls far below 100%.

JET–CONDENSING APPARATUS

Several of the above statements relative to surface condenser practice apply with equal force to jet-condensing arrangements, so that there is little to add under this heading.

The auxiliaries required for the operation of a jet condenser are — (1), an air-pump; and (2), an injection water-pump.   In cases where water of sufficient pressure is available, as in a pumping plant, or where water under city pressure is used for injection, the latter pump is not required.

AIR-PUMP

The power required to drive the air-pump is dependent upon the quantity of injection water used, the air leakage into the system, and the head against which the water is discharged.   The proper quantity of injection water varies between 20 and 50 lbs. for each pound of exhaust steam, depending upon its temperature and the vacuum to be obtained.   This water places the largest duty upon the air-pump.

Contrary to surface condenser practice, the air leakage, while a serious impediment to high vacuum, is regarded as a load upon the air-pump, of comparatively slight importance, for the reason that the large amount of water present is sufficient to absorb the greater part of the air, rendering its handling a much more simple problem.

The head discharged against is usually just sufficient to pump the heated injection water and condensed steam to waste or reservoir and rarely exceeds five or ten feet, to which must be added, however, the equivalent of the vacuum maintained against which the suction of the pump must work. The efficiency of horizontal single-direct acting independent air pumps ranges between 75 and 85%, while the steam required per indicated horsepower per hour should be between 80 and 90 lbs. for larger compound pumps well lagged and up to 150 lbs. for smaller simple steam cylinder arrangements.

ILLUSTRATION

To illustrate the above figures, the following examples may be of interest:

	Surface Cond. Plant	Jet. Cond. Plant
Size of plant....................	1000 I. H. P.	1500 I. H. P.
Steam consumption of main engine per I. H. P. per hour..........	15 pounds	14 pounds
Cooling water used per lb. exhaust steam......................	40 pounds	30 pounds
Head against which cooling water is discharged..................	20 feet	40 feet
Kind of air-pump used..........	Vertical crank and fly-wheel, steam driven	Horizontal single D. A. steam driven
Kind of circulating pumps used ...	Centrifugal D. C. to engine	None

In the above two examples, the total steam per hour required by the condenser auxiliaries, assuming sufficient pressure available to deliver injection water in the case of the jet-condensing plant may be calculated as follows:

Starting with the surface-condensing plant, we will assume the following:

Steam consumption of air-pump per I. H. P. per hour................ 50 lbs.
Efficiency of air-pump.. 60%
Steam consumption of circulating pump engine per I. H. P. per hour ... 40 lbs.
Combined efficiency of centrifugal pump and engine.................. 50%

With the above data complete, all necessary information is at hand for determining the required amounts. The total steam used per hour by the main engine will equal 1000 × 15 or 15,000 lbs. of steam, which, multiplied by 40, will give 600,000 lbs. circulating water used per hour. From this,

$$\frac{600,000 \times 20}{33,000 \times 60} = 6.06 \text{ H. P.}$$

actually required to pump the circulating water. This is equivalent, therefore, with the efficiency of 50% above assumed, to

$$\frac{6.06}{.5} = 12.12 \text{ I. H. P.}$$

in the steam cylinder of the circulating pump engine. But this auxiliary engine requires 40 lbs. steam per I. H. P. per hour. Therefore,

$$12.12 \times 40 = 484.8 \text{ lbs.}$$

total steam required per hour by circulating pump. For the air-pump,

$$\frac{.00024 \times 15000 \times 50}{.6} = 300 \text{ lbs.}$$

total steam required per hour.   From which the steam required by both pumps per hour is equivalent to

$$484.8 + 300 \text{ or } 784.8 \text{ lbs.}$$

In the jet-condenser plant, we will assume:

Steam consumption of air-pump per I. H. P. hour . . . .   100 lbs.
Efficiency of air-pump 80%

The total quantity of injection water will be

$$1500 \times 14 \times 30 = 630{,}000 \text{ lbs.}$$

per hour, and the steam required to pump it, together with the condensed steam will be

$$\frac{(630{,}000 + 21{,}000) \times 40 \times 100}{33{,}000 \times 60 \times .8}$$

or 1644 lbs. total steam per hour.

The above example is given at this place so that the method employed in expressing the steam consumption of the auxiliaries in the more complicated formulæ to follow will be more easily understood.

FEED–PUMPS

It is now considered the best practice to install duplex direct-acting feed-pumps for reasons of convenience in handling, regulation, and general flexibility, even though the air and circulating pumps be motor or belt driven.   The steam consumption of these appliances varies considerably with the size selected and the working conditions involved.

With compound steam ends well lagged and covered, 100 lbs. of steam per I. H. P. per hour should be safe for large sizes, while in small pumps, 200 lbs. appears to be nearer the mark.   The pump efficiency should not be less than 80%.

The steam consumption and efficiency of the feed-pump being known, it is only necessary to obtain the head pumped against (*i.e.*, the boiler pressure), and the quantity of water pumped in order to obtain the steam used by the feed-pump per hour.

The former is known beforehand for any special case; the latter must be calculated.   Suppose in a given plant there is required by the main engine, oil burners, radiation and leakage and all the auxiliaries with the

exception of the feed-pump, 10,000 lbs. of steam per hour when operating at full rated load, and the boiler pressure is 150 lbs. by gage. The feed-pump must now pump not only the 10,000 lbs. of water against 150 lbs. pressure, but, in addition, the actual amount of steam required to operate the feed-pump itself. Assuming the economy of the pump to be 200 lbs. of steam per I. H. P. per hour and its efficiency to be 80%, and letting "p" represent the total steam used by the feed-pump per hour, the above idea is algebraically represented as follows:

$$p = \frac{(10,000 + p) \times 150 \times 2.31 \times 200}{33,000 \times 60 \times .8}$$

whence the value of "p" may be easily obtained by solving the equation. The exact manner in which this idea is applied to complete plant calculations will be shown later.

OIL PUMPS

SMALL duplex oil pumps of the type usually selected for feeding oil to the burners are exceedingly uneconomical and a steam consumption of 200 lbs. of steam per I. H. P. per hour may be considered a fair average. The pump efficiency also drops exceedingly low, due to the heavy viscous nature of the fluid pumped, the uncertainty — not to say impossibility — of filling the cylinders with oil on each stroke, and the constant presence of a varying proportion of gas. This efficiency therefore varies from 40 to 50%, and is sometimes much lower. The exhaust from the oil pumps is usually employed to heat the oil after being discharged from the pumps, but before reaching the burners.

Assuming 60 lbs. oil pressure in the discharge line, a pump efficiency of 40%, a steam consumption of 250 lbs. per I. H. P. per hour, an evaporation in the boiler of 12 lbs. of water per pound of oil, and also that 50% excess oil is by-passed through the oil relief valve to the storage tank, the steam required by the oil pump for each 12 lbs. of water evaporated, would be

$$\frac{3 \times 60 \times 2.31 \times 250}{2 \times 60 \times 33,000 \times .4}$$

or .066 lbs. of steam. Expressed as a percentage of the total evaporation in the boiler, this would be equivalent to

$$\frac{.066}{12} = .0055$$

which is less than 1%. This method of figuring is subject to considerable error, owing to variations between the actual and assumed conditions. It is therefore customary to directly assume the steam consumption of

the oil pumps as being 1% of the total evaporation, which is conservative. Subsequent figures will therefore be based upon this assumption.

OIL BURNERS, RADIATION AND LEAKAGE

INASMUCH as the steam used for atomizing the oil at the burners is a direct loss, the quantity required is of considerable importance. This depends somewhat upon the burner and furnace arrangement, and upon the skill of the operator. Tests have been made showing less than 2% of the total evaporation as being sufficient steam with some burners for this pupose. This requires, however, close adjustment and careful attention. Under favorable conditions, 3% should be amply conservative, and this figure will be used in the analyses to follow.

Radiation and leakage from piping, valves, auxiliaries, etc., is also a large factor in the resultant economy. This does not include radiation from the boiler or main engine as these losses are included in the assumed efficiencies. In a well-designed plant with a good piping arrangement, carefully covered with 85% carbonate of magnesia, this loss should not exceed from 2 to $2\frac{1}{2}$%. This would not hold true if applied to the many mongrel plants, containing a wonderful selection of poorly designed and built machines of miscellaneous types abominably installed, which we see on every hand. In the following figures, however, 3% will be taken to represent this loss in all cases.

While on this subject, it might be appropriate to mention the fact that with steam of 100° superheat, the radiation loss would appear less than with saturated steam despite the higher temperature. This looks paradoxical, but it is nevertheless a fact. Several theories have been advanced of late to account for this phenomenon, the most reasonable among which is that with saturated steam a film of water of condensation adheres to the inside surface of the pipe, which rapidly acts as a transmitting medium between the steam and the pipe, thus assisting radiation. With superheated steam, on the other hand, no moisture is present, and the steam, itself, being a non-conductor, very little radiation results. Be this as it may, there is very little doubt but that radiation is less with superheated than with saturated steam.

Adequate experiments have not, up to the present time, been made under the actual working conditions of steam-power plants to enable one to state with absolute assurance just exactly what the radiation and leakage loss is, but the assumptions above made are believed to be very nearly correct for the higher grade of plant.

FEED-WATER HEATERS

A FEED-WATER heater is, as its name implies, a piece of apparatus for heating the feed-water before it enters the boilers.

Types of Heaters

There are two general classes, viz.: the open type and the closed type. In the former, the exhaust steam which is used as a heating medium in both types, is brought into direct contact with the water to be heated; in the latter type, the water is separated from the steam by one of several forms of heating surface. So long as this heating surface remains clean and its conductivity thereby maintained at a maximum, the temperature of the water may be raised to practically that of the exhaust steam or 212° Fahr. Under these conditions, there is very little difference between the efficiencies of the two types of heaters, the actual difference resulting from the fact that with the open type heater, the steam used for heating condenses and forms part of the feed-water, thus requiring a slightly smaller quantity of exhaust steam to obtain the same final temperature of the feed. Should the water side of the heating surface of a closed type heater become coated with scale and the steam side with a film of cylinder oil, the conductivity and therefore the efficiency is greatly reduced.

On the other hand, the feed-water from an open type heater is composed partly of condensed steam and partly of heated water from an outside source, so that should the cylinder oil contained in that part of the feed-water formed by the condensation of the exhaust steam find its way into the boilers, forming a coating of scale upon the heating surface, the efficiency of the boilers would be appreciably reduced, ultimately resulting in the burning out of tubes.

Thus it is, that should a closed type heater be selected, provision should be made for periodically cleaning its heating surface, while if an open type heater be chosen, an efficient means of eliminating the cylinder oil from the exhaust steam before it enters the feed-water heater, or from the condensed steam, before it enters the boilers, should be provided.

Economy of Feed-Water Heaters

The advantages to be derived from the use of feed-water heaters result from the utilization of the otherwise waste heat contained in the exhaust steam from the main engine or auxiliaries. In a non-condensing plant, the exhaust steam from the main engine may be utilized, while for condensing plants the heat in the auxiliary exhaust may profitably be abstracted.

As an example of the saving to be effected in this manner, let us consider the case of a non-condensing plant, assuming the exhaust from the engines under atmospheric pressure to be utilized for heating the feed-water. Under these conditions, each pound of exhaust steam will be capable of liberating about 966 B. T. U. without diminishing its own temperature. In other words, the latent heat of steam at atmospheric

pressure is about 966 B. T. U.   This quantity of heat is sufficient to raise the temperature of one pound of water 966° Fahr., or to raise the temperature of 966 pounds of water 1° Fahr., which is equivalent to raising the temperature of 6 lbs. of water 161° Fahr.   With water having an initial temperature of 50° Fahr., the resulting temperature of a mixture of 6 lbs. of water with 1 lb. of exhaust steam will be 211° Fahr., or practically boiling point.   The exhaust steam from the engine represents approximately $\frac{5}{6}$ of the total quantity of steam generated in the boilers, the remaining $\frac{1}{6}$ being utilized by the auxiliaries, etc.

It therefore follows that $\frac{1}{6} \times \frac{5}{6}$ or $\frac{1}{7}$ of the exhaust steam from the engine will heat the total quantity of feed-water from 50° Fahr. to the boiling point in the case of a closed heater.   In an open heater, the steam used for heating the water actually mixes with it so that the result of mixing one pound of exhaust steam with six pounds of water at 50° Fahr., is seven pounds of water at the boiling point.   In this case,

$$\frac{1}{7} \times \frac{6}{5} \text{ or } \frac{6}{35} - \text{ say roughly } \frac{1}{6}$$

of the exhaust steam from the engine is sufficient to heat all of the feed-water to the boiling point.

Assuming a boiler pressure by gage of 150 lbs., each pound of steam will contain 1191.2 B. T. U. above 32° Fahr.   One pound of water at 50° Fahr. contains 18.1 B. T. U.   Due to the action of the feed-water heater, there has, therefore, been added to each pound of water pumped to the boiler, the difference between the heat it contained at 50° Fahr., and that at 211° Fahr. or

$$179.78 - 18.1 = 161.68 \text{ B. T. U.}$$

which is the net gain effected by the heater.   Expressed as a percentage, this is equivalent to

$$\frac{161.68}{1191.2 - 18.1} \text{ or about } 14\%.$$

From this it will be seen that a raise of 160° in the temperature of the feed-water is equivalent to a saving of 14%, or for every 11.4° raise in the temperature of the feed-water, 1% of the annual fuel bill is saved.

ECONOMIZERS

"ECONOMIZER" is a technical term designating a type of feed-water heater in which the waste gases from the boiler are used as a heating medium in lieu of the exhaust steam from the feed-pumps.   At first thought it would appear that owing to the comparatively high temperature of these gases, a saving in economy might be realized by the use of an economizer, compensurate with this temperature, but when it is remembered

that the specific heat of the flue gases is less than one-quarter that of the water to be heated, and that accordingly the loss in their temperature is more than four times the rise in temperature of the feed-water, the fallacy of such a proposition is readily seen.

EFFICIENCY OF ECONOMIZERS

It is a fact, however, that the fuel economizer properly applied exercises a decided economical effect upon the ultimate economy of a steam-power plant, its efficiency depending upon the following conditions:

(1) The quantity of flue gases available;

(2) Their temperature;

(3) The temperature of the entering water;

(4) The quantity of heating surface;

(5) Physical condition of the heating surface.

THE FLUE GASES

The quantity and temperature of the flue gases, always assuming a water-tube boiler installation of high grade, depend upon the degree of excellence of the furnace equipment and the skill of the fireman. But inasmuch as the present remarks refer to continuous full load operation under test conditions, the latter element may be neglected. Results of numerous tests show that when operating at maximum economy with furnaces carefully adjusted, the temperature of the flue gases should be not exceeding 440° Fahr., and the excess air supply not over 25% above theoretical chemical requirements.

FEED–WATER

The efficiency of the economizer is much greater with low temperatures of entering feed-water than in cases where this temperature is comparatively high. Hence, in order to obtain the best results, the air and circulating pumps should be either belted or direct connected to motors and operated by current from the main units. In this way, no exhaust steam is left to wastefully heat the feed-water, the auxiliaries are in themselves economical, the economizer efficiency is increased, and the final power-plant economy correspondingly improved. It is considered the better practice to still maintain steam-driven feed-pumps, however, and a feed-water heater is therefore employed to utilize the resultant exhaust steam.

HEATING SURFACE

The quantity of heating surface installed for Pacific Coast conditions varies between two and four boiler horse-power per economizer tube, each tube being under standard conditions ten feet long and containing

about twelve square feet of heating surface.   In predetermining the exact amount of heating surface advisable for a proposed installation, the probable. rise in temperature for different quantities should be estimated, and the saving thus effected, expressed as an annual interest upon the cost of the economizer installation, whence the most attractive proposition will immediately appear.

The same may be said regarding the conditions of the heating surface with reference to the skill of the fireman.   The test conditions necessarily presuppose all apparatus to be in first-class working order.

From the above remarks it will be evident that in general a fuel economizer is far less effective with oil than with coal as fuel.   The following comparisons may be of interest in this connection:

With oil, the temperature of the flue gases is lowered, the required air supply is less, and the boiler efficiency is higher than with coal, other things being equal.

Due to the comparatively slight draft required for oil burning, forced or induced draft is never necessitated by the reduction of the stack temperature even with a stack only sixty feet high.   The loss attendant upon such apparatus is, therefore, not to be figured in an oil-burning plant.   Further, practically no soot is formed, so that the economizer scraping gear need be used only during a few hours per day.

PART II

THE FACTOR OF EVAPORATION

HAVING briefly reviewed the leading principles governing the use of feed-water heaters and fuel economizers, we are at once compelled to devote a few words to that element, which effected largely by this temperature, exercises a correspondingly great influence upon the net economy of any steam plant, *i.e.*, the factor of evaporation. In fact, as will be shown in the following pages, the plant economy as expressed in kilowatt hours per barrel of oil varies inversely with this factor.

In order to ascertain the numerical value of this element, it is necessary to know the pressure and temperature of the steam as generated and the temperature of the entering feed-water.

It will be apparent that the higher the temperature of the water when entering the boiler and the lower the pressure, and hence the temperature of the steam, the less will be the work imposed upon the boiler and vice versa. The factor of evaporation simply reduces the performance of any boiler under any conditions of steam pressure and feed-water temperature to a comparative basis of "from and at 212° Fahr." That is, the actual evaporation obtained in a given boiler when multiplied by this factor, gives the equivalent evaporation which would have been attained were the temperature of the feed-water 212° Fahr. and the actual boiler pressure one atmosphere.

Expressed as a formula, the factor of evaporation equals

$$F = \frac{H - h}{965.7}$$

in which

F = factor of evaporation;

H = the heat contained in one pound of steam above 32° Fahr.;

h = the heat contained in one pound of water above 32° Fahr.;

965.7 = latent heat of steam at atmospheric pressure.

"H" is readily found from any of the tables showing the properties of saturated steam; but in order to find "h," the temperature of the feed must be known.

TEMPERATURE OF THE FEED-WATER

In order to ascertain this value, Figs. 14 to 18 will be found useful. Figure 14 shows the results to be obtained from both open and closed heaters for various temperatures of air-pump discharge and percentages of exhaust steam available. Condensing plants are assumed throughout.
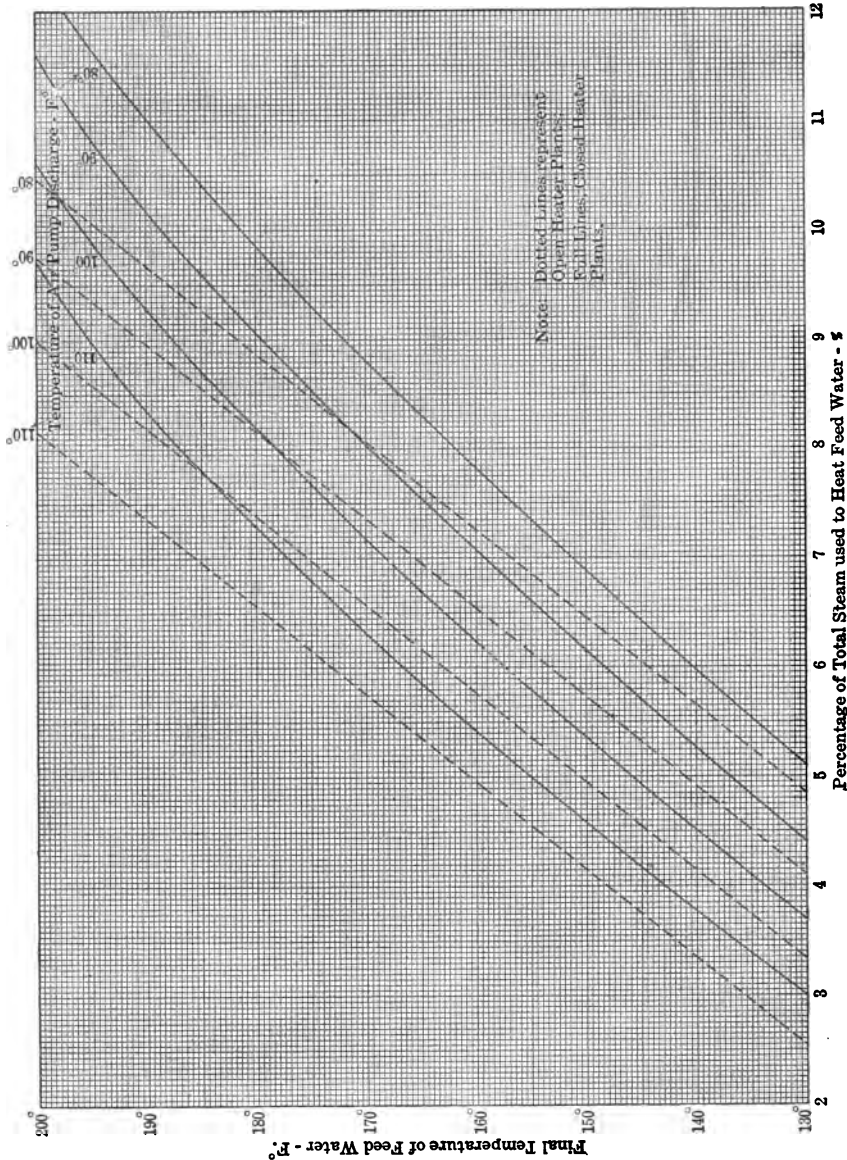
FIG. 14. — APPROXIMATE FEED-WATER TEMPERATURES — CONDENSING PLANTS

For non-condensing plants more than sufficient steam is always available to obtain the maximum temperature of 212° Fahr.

Figure 15 sets forth the same data but also shows the effect of fuel economizers for various amounts of economizer heating surface. To illustrate the use of this graph assume a plant containing an open type heater, the temperature of the air-pump discharge 100° Fahr., 4% of the
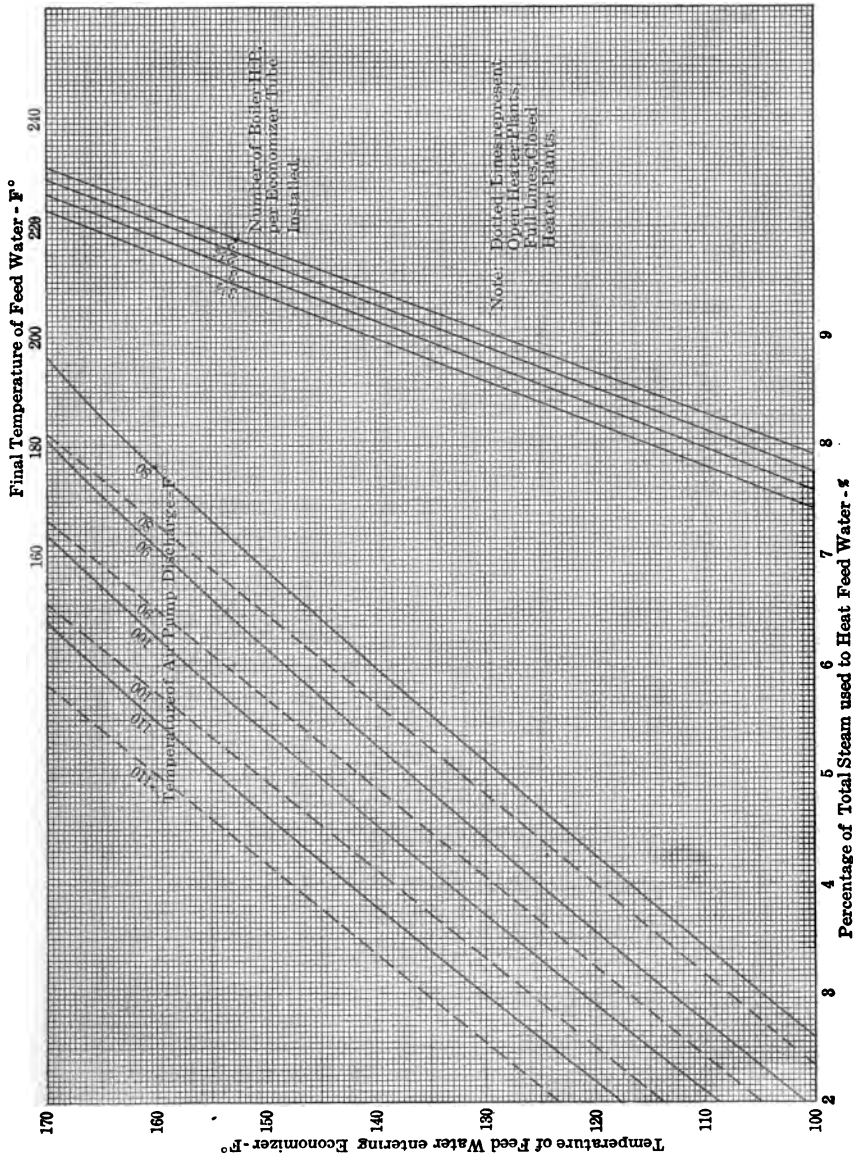
Fig. 15. — Approximate Feed-Water Temperatures — Condensing Plants with Economizers

total steam generated used by the feed-pumps and afterwards made available for heating the feed-water in the open type heater above referred to,

FIG 16. — PERCENTAGE OF STEAM GENERATED USED BY AUXILIARIES
(SURFACE-CONDENSING PLANTS — STEAM-DRIVEN PUMPS)

KIND OF PLANT	FEED-PUMP		CIRC. PUMP		AIR-PUMP		"Wh."	Steam Used by Auxiliaries %
	Economy	Efficiency	Economy	Efficiency	Economy	Efficiency		
75 to 500 K.W. Horizontal — direct-acting auxiliaries	200 lbs.	80%	150 lbs.	80%	150 lbs.	60%	500 1000 1500 2000 2500	12.50 15.75 19.00 22.00 24.75
500 to 1000 K.W. Horizontal — direct-acting auxiliaries	150 lbs.	80%	100 lbs.	80%	100 lbs.	60%	500 1000 1500 2000 2500	9.00 11.50 13.75 16.00 18.25
100 to 600 K.W. Engine-driven centrifugal circulating pumps— crank and fly-wheel air-pumps — direct-acting feed-pumps	200 lbs.	80%	60 lbs.	50%	50 lbs.	60%	500 1000 1500 2000 2500	8.75 11.00 13.00 15.25 17.25
600 to 1000 K.W. Engine-driven centrifugal circulating pumps— crank and fly-wheel air-pumps — direct-acting feed-pumps	150 lbs.	80%	50 lbs.	52%	45 lbs.	60%	500 1000 1500 2000 2500	7.50 9.50 11.50 13.50 15.50
Above 1000 K.W. Engine-driven centrifugal circulating pumps— crank and fly-weeel air-pumps — direct-acting feed pumps	100 lbs.	80%	40 lbs.	55%	40 lbs.	60%	500 1000 1500 2000 2500	5.25 6.75 8.25 10.00 11.50

and lastly an economizer containing heating surface equivalent to two boiler horse-power per tube.

To find the feed-water temperature in this plant under full load test conditions, locate 4% on the lower horizontal scale of percentages and move vertically until the dotted line showing the air-pump discharge

to be 100° Fahr., is intersected. If there were no economizer, the temperature would now be 138.5°, but the water, instead of entering the boiler, reaches the economizer at this temperature. From the point of intersection thus found, move horizontally until reaching the heavy line denoting two boiler horse-power per economizer tube, when the final temperature of the feed-water will be found immediately above, at the top of the chart. In this case it is about 207.5° Fahr.

Care should be exercised both in Figs. 14 and 15 in selecting proper values for the temperature of the hot-well or air-pump discharge. This

FIG. 17. — PERCENTAGE OF STEAM GENERATED USED BY FEED-PUMP
(SURFACE-CONDENSING PLANTS — POWER-DRIVEN AUXILIARIES)

KIND OF PLANT	FEED-PUMP		Combined Efficiency of Circulating Pump	Combined Efficiency of Air-Pump	"Wh."	Steam Used by Feed-Pump %
	Economy	Efficiency				
100 to 600 K.W. Power-driven air and circulating pumps—direct-acting feed-pumps.	200 lbs.	80%	36%	40%	500	4.50
					1000	4.50
					1500	4.50
					2000	4.50
					2500	4.50
600 to 1000 K.W. Power-driven air and circulating pumps—direct-acting feed-pumps.	150 lbs.	80%	43%	44%	500	3.25
					1000	3.25
					1500	3.25
					2000	3.25
					2500	3.25
Above 1000 K.W. Power-driven air and circulating pumps—direct-acting feed-pumps.	100 lbs.	80%	48%	48%	500	2.00
					1000	2.00
					1500	2.00
					2000	2.00
					2500	2.00

depends upon the vacuum maintained and the character of condensing apparatus. In surface-condensing plants, where high vacuums are maintained and the cooling water is at a low initial temperature, the air pump discharge is usually from 15 to 20° Fahr. below the temperature of the exhaust steam under vacuum. To counteract this loss, a primary heater is sometimes used in high-grade plants, consisting of a proper amount of heating surface in the top of the condenser shell through which the air-pump discharge is pumped, thus being reheated by the exhaust from the main engine to within three or five degrees of its own temperature.

Both Figs. 14 and 15 may be used for jet as well as surface-condensing plants, provided the feed-water is at a temperature of not less than 80°

Fahr. before entering the heater.   In Fig. 15., the economizer curves are plotted to correspond to the best boiler performance, *i.e.*, 440° flue gases, and 25% excess air supply; the boiler falling below this standard, the econo-

FIG. 18. — PERCENTAGE OF STEAM GENERATED USED BY AUXILIARIES
(JET-CONDENSING PLANTS—STEAM-DRIVEN PUMPS)

KIND OF PLANT	FEED-PUMP		INJECTION PUMP		AIR-PUMP		"Wh"	Steam Used by Auxiliaries %
	Economy	Efficiency	Economy	Efficiency	Economy	Efficiency		
50 to 300 K.W. Horizontal—direct-acting auxiliaries.	200 lbs.	80%	150 lbs.	80%	150 lbs.	70%	200 300 500 1000 1500	13.00 16.75 20.75 25.75 30.00
300 to 800 K.W. Horizontal—direct-acting auxiliaries.	150 lbs.	80%	100 lbs.	80%	100 lbs.	70%	200 300 500 1000 1500	9.00 11.50 14.75 18.50 25.00
150 to 500 K.W. Engine-driven centrifugal injection pumps—crank and fly-wheel air-pumps — direct-acting feed-pumps.	200 lbs.	80%	60 lbs.	50%	50 lbs.	70%	200 300 500 1000 1500	8.00 9.50 11.50 14.75 19.75
500 to 1000 K.W. Engine-driven centrifugal injection pumps—crank and fly-wheel air-pumps — direct-acting feed-pumps.	150 lbs.	80%	50 lbs.	52%	45 lbs.	70%	200 300 500 1000 1500	6.00 7.50 9.25 12.00 16.75
Above 1000 K.W. Engine-driven centrifugal injection pumps—crank and fly-wheel air-pumps — direct-acting feed-pumps.	100 lbs.	80%	40 lbs.	55%	40 lbs.	70%	200 300 500 1000 1500	4.50 5.75 7.25 9.75 13.75

mizer results will be better than shown.   This is one of the strongest merits of the fuel economizer, viz.: that when the boiler performance decreases, due to careless firing, the efficiency of the economizer increases.

Steam Used by Auxiliaries

In order to obtain the percentages of steam used by the auxiliaries, the exhaust from which is utilized in heating the feed-water, Figs. 16, 17, and 18 may be used.

Figure 16 is for surface-condensing plants having steam-driven auxiliaries, the last column containing the total amount of steam used by

Fig. 19. — Factors of Evaporation — Saturated Steam

Temperature of Feed F°	Steam Pressure by Gage — Lbs.										
	100	110	120	130	140	150	160	170	180	190	200
80	1.177	1.179	1.181	1.183	1.184	1.186	1.187	1.189	1.190	1.192	1.193
90	1.167	1.169	1.170	1.172	1.174	1.176	1.177	1.179	1.180	1.181	1.183
100	1.156	1.158	1.160	1.162	1.164	1.165	1.167	1.168	1.170	1.171	1.172
110	1.146	1.148	1.150	1.152	1.153	1.155	1.156	1.158	1.159	1.160	1.162
120	1.136	1.138	1.140	1.141	1.143	1.145	1.146	1.147	1.149	1.150	1.151
130	1.125	1.127	1.129	1.130	1.132	1.134	1.136	1.137	1.138	1.140	1.141
140	1.115	1.117	1.119	1.120	1.122	1.124	1.125	1.127	1.128	1.129	1.131
150	1.104	1.106	1.108	1.110	1.111	1.113	1.115	1.116	1.118	1.119	1.120
160	1.094	1.096	1.098	1.100	1.101	1.103	1.104	1.106	1.107	1.108	1.110
170	1.083	1.085	1.087	1.089	1.091	1.092	1.094	1.095	1.097	1.098	1.099
180	1.073	1.075	1.077	1.079	1.080	1.082	1.083	1.085	1.086	1.088	1.089
190	1.063	1.065	1.066	1.068	1.070	1.071	1.073	1.074	1.076	1.077	1.078
200	1.052	1.054	1.056	1.058	1.059	1.061	1.063	1.064	1.065	1.067	1.068
210	1.042	1.044	1.046	1.047	1.049	1.051	1.052	1.053	1.055	1.056	1.057
220	1.031	1.033	1.035	1.037	1.039	1.040	1.042	1.043	1.045	1.046	1.047
230	1.021	1.023	1.025	1.027	1.028	1.030	1.031	1.033	1.034	1.035	1.037
240	1.011	1.013	1.014	1.016	1.018	1.019	1.021	1.022	1.024	1.025	1.026
250	1.000	1.002	1.004	1.006	1.007	1.009	1.010	1.012	1.013	1.015	1.016
260	0.990	0.992	0.993	0.995	0.997	0.999	1.000	1.001	1.003	1.004	1.005
270	0.979	0.981	0.983	0.985	0.986	0.988	0.990	0.991	0.992	0.994	0.995
280	0.969	0.971	0.973	0.974	0.976	0.978	0.979	0.981	0.982	0.983	0.985
290	0.958	0.960	0.962	0.964	0.966	0.967	0.969	0.970	0.972	0.973	0.974
300	0.949	0.950	0.952	0.953	0.955	0.957	0.958	0.960	0.961	0.962	0.964

the feed, air, and circulating pumps expressed as a percentage of the total evaporation in the boiler.

Figure 17 represents the percentage of steam used by the feed-pumps in surface-condensing plants having power-driven air and circulating pumps. The "combined efficiency" of air and circulating pumps represent the product of the pump efficiency with its respective motor efficiency, or belt drive efficiency. This figure will be found useful in economizer plants.

Figure 18 sets forth the same information for jet-condensing plants. The last three tables give the percentages necessary to apply Figs. 14 and 15. The calculations to follow will show clearly the manner in which these values have been calculated.

Having obtained the proper temperatures for the feed-water, and knowing the boiler pressure, the factors of evaporation may be read di-

FIG. 20. — FACTORS OF EVAPORATION.   (SUPERHEATED STEAM — 120° F.)

Temeprature of Feed F°	STEAM PRESSURE BY GAGE — LBS.										
	100	110	120	130	140	150	160	170	180	190	200
80	1.258	1.260	1.262	1.264	1.265	1.267	1.268	1.270	1.271	1.273	1.274
90	1.248	1.250	1.251	1.253	1.255	1.257	1.258	1.260	1.261	1.262	1.264
100	1.237	1.239	1.241	1.243	1.245	1.246	1.248	1.249	1.251	1.252	1.253
110	1.227	1.229	1.231	1.233	1.234	1.236	1.237	1.239	1.240	1.241	1.243
120	1.217	1.219	1.221	1.222	1.224	1.226	1.227	1.228	1.230	1.231	1.232
130	1.206	1.208	1.210	1.211	1.213	1.215	1.217	1.218	1.219	1.221	1.222
140	1.196	1.198	1.200	1.201	1.203	1.205	1.206	1.208	1.209	1.210	1.212
150	1.185	1.187	1.189	1.191	1.192	1.194	1.196	1.197	1.199	1.200	1.201
160	1.175	1.177	1.179	1.181	1.182	1.184	1.185	1.187	1.188	1.189	1.191
170	1.164	1.166	1.168	1.170	1.172	1.173	1.175	1.176	1.178	1.179	1.180
180	1.154	1.156	1.158	1.160	1.161	1.163	1.164	1.166	1.167	1.169	1.170
190	1.144	1.146	1.147	1.149	1.151	1.152	1.154	1.155	1.157	1.158	1.159
200	1.133	1.135	1.137	1.139	1.140	1.142	1.144	1.145	1.146	1.148	1.149
210	1.123	1.125	1.127	1.128	1.130	1.132	1.133	1.134	1.136	1.137	1.138
220	1.112	1.114	1.116	1.118	1.120	1.121	1.123	1.124	1.126	1.127	1.128
230	1.102	1.104	1.106	1.108	1.109	1.111	1.112	1.114	1.115	1.116	1.118
240	1.092	1.094	1.095	1.097	1.099	1.100	1.102	1.103	1.105	1.106	1.107
250	1.081	1.083	1.085	1.087	1.088	1.090	1.091	1.093	1.094	1.096	1.097
260	1.071	1.073	1.074	1.076	1.078	1.080	1.081	1.082	1.084	1.085	1.086
270	1.060	1.062	1.064	1.066	1.067	1.069	1.071	1.072	1.073	1.075	1.076
280	1.050	1.052	1.054	1.055	1.057	1.059	1.060	1.062	1.063	1.064	1.066
290	1.039	1.041	1.043	1.045	1.047	1.048	1.050	1.051	1.053	1.054	1.055
300	1.030	1.031	1.033	1.034	1.036	1.038	1.039	1.041	1.042	1.043	1.045

rectly from Fig. 19, for saturated steam and from Fig. 20, for steam of 120° Fahr. superheat.   In the latter, the mean value of .65 has been taken as the specific heat of the steam.

PART III

COMPLETE PLANT ECONOMY

(FULL RATED LOAD)

HAVING briefly investigated the economies of the different units entering into a steam-power plant, and the conditions which tend to vary these economies, a method will now be developed by means of which these results can be combined, thus obtaining the net resultant economy of a complete plant. This will be expressed in terms of kilowatt hours output at the switchboard per barrel of oil burned.

It should be remembered, however, that the method as outlined in the following pages may be applied with equal facility to power plants burning wood, coal, or any other fuel. In order, therefore, to convert the results as derived from the various formulæ into equivalent figures per ton of any other fuel, the conversion table, Fig. 20-A, is given. The process, then, in solving for — say a coal-burning plant is as follows:

Apply the proper formula as shown hereinafter for the type of plant under consideration, using either of the three values given for the heat value of crude oil, and being careful to substitute for the boiler efficiency a percentage that may be attained with the kind of coal and furnace arrangement to be used. This result will represent Kw. hours per 336 lbs. (one barrel) of coal having the heat value used in solving the equation. To correct for this heat value and for the steam used in operating the oil-pumps and burners (which is not required in any plant burning solid fuel), Fig. 20-A may be used.

Locating the result of the formula upon the right-hand margin and moving horizontally until intersecting one of the three lines representing the calorific value used in applying the formula, the result in terms of Kw. hours per ton (2000 lbs.) of coal may be read directly below upon the lower scale, assuming the coal to contain 10,000 B. T. U. per pound. For any other heat value of the coal, move vertically from the intersection above found to the line representing the required value, thence horizontally to the left-hand scale where the proper result will be found. The Kw. hours per 100 lbs. coal will, of course, be one-twentieth of this result in all cases.

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Figure 20-B is self-explanatory, and shows the relation between the Kw. hours per barrel oil and the pounds oil required both per Kw. hour and per electrical H. P. hour.   Similarly, Fig. 20-C shows the relation between Kw. hours per ton of coal or other fuel and the pounds fuel per Kw. hour and per electrical H. P. hour.
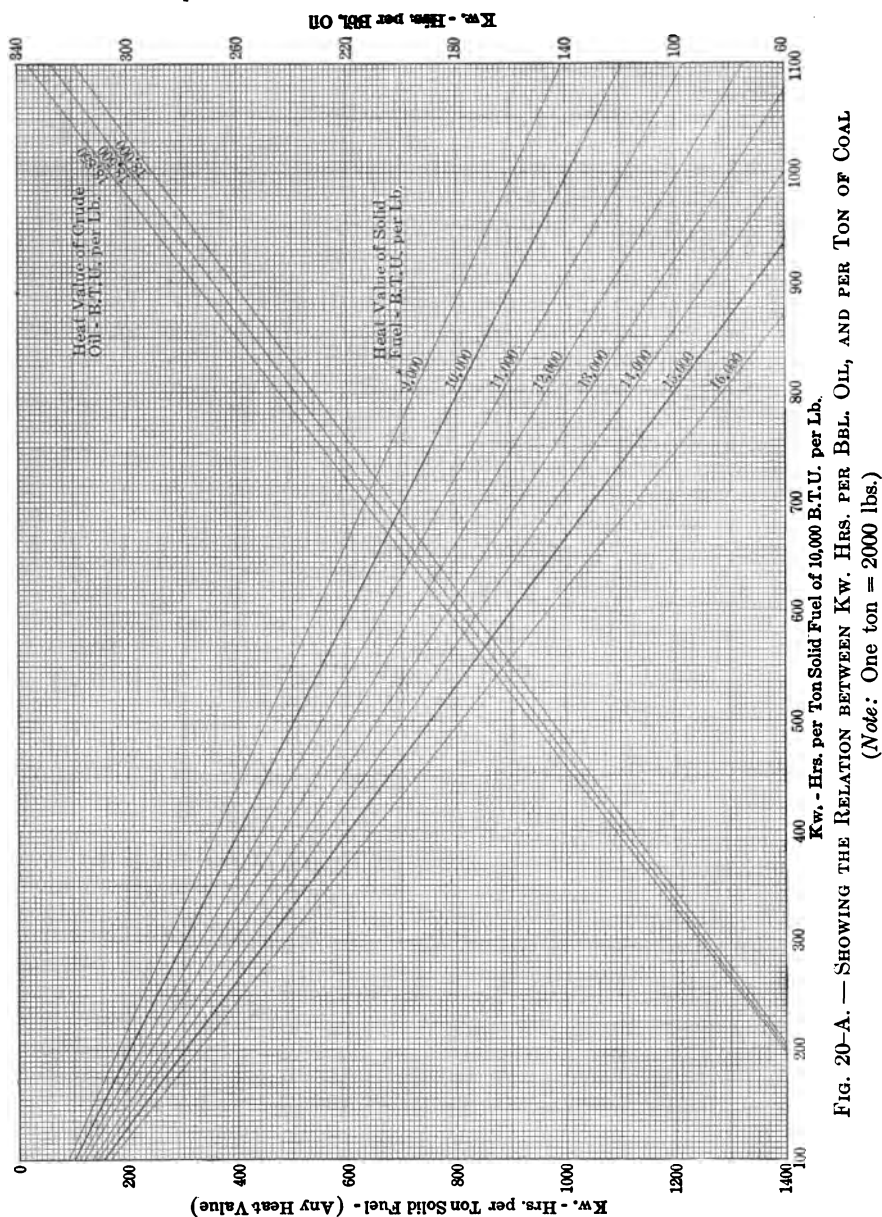
FIG. 20-A. — SHOWING THE RELATION BETWEEN KW. HRS. PER BBL. OIL, AND PER TON OF COAL

(*Note:* One ton = 2000 lbs.)

As a matter of interest, the best result to be expected from the highest grade steam power-plant having superheaters and other economical devices when burning crude oil, is in the neighborhood of 260 Kw. hours per barrel, which as shown by Fig. 20-B is equivalent to slightly under one pound of oil per electrical H. P.
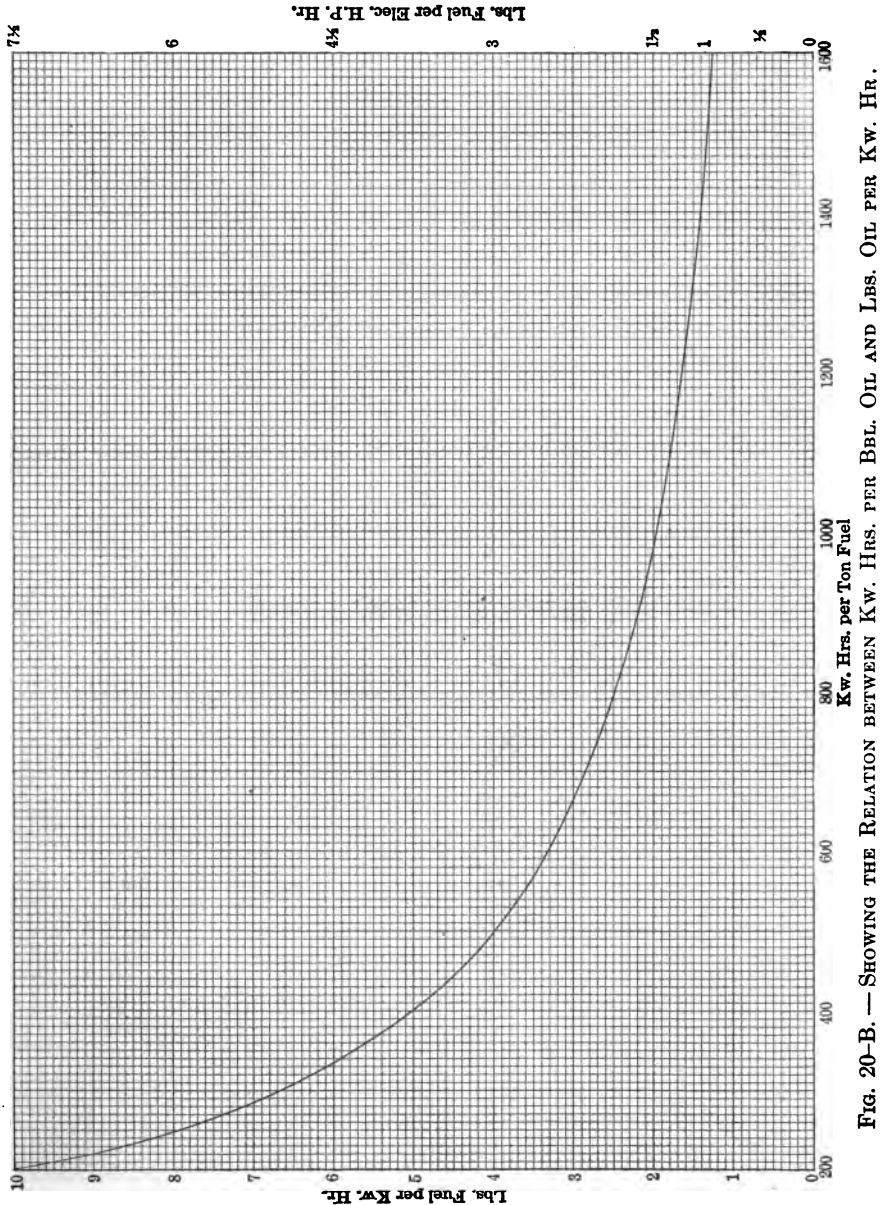
FIG. 20-B. — SHOWING THE RELATION BETWEEN KW. HRS. PER BBL. OIL AND LBS. OIL PER KW. HR.

In comparing oil with coal values Fig. 20-A, as stated above, takes into consideration the 4% of steam mentioned hereinafter as being necessary for the oil-pumps and burners. The figure therefore assumes hand firing, and if mechanical stokers are used, or any other apparatus requiring steam and therefore fuel, for its operation, results as derived from
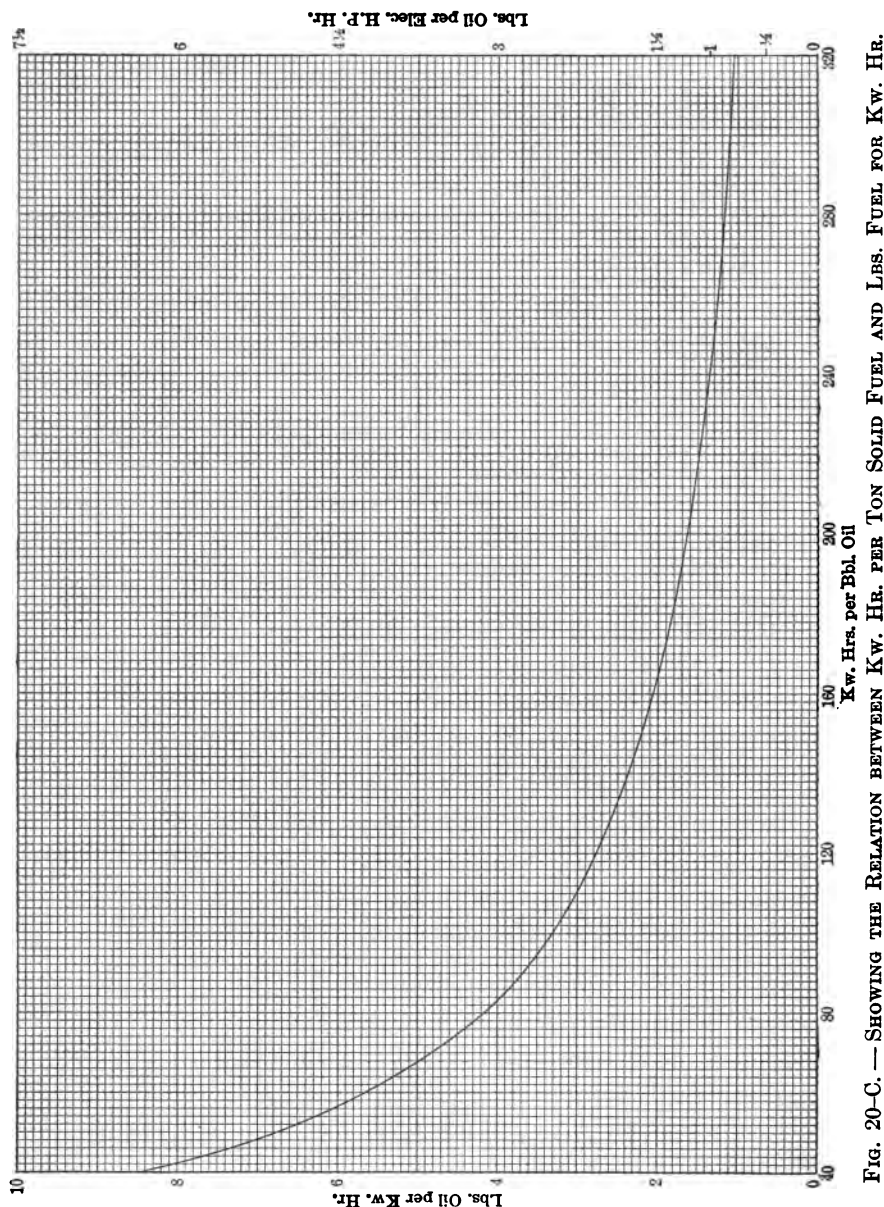
Fig. 20-C. — Showing the Relation between Kw. Hr. per Ton Solid Fuel and Lbs. Fuel for Kw. Hr.

Fig. 20-A should be corrected accordingly. This figure is of special interest in showing the economy for oil equivalent to the known performance of any particular plant burning coal or wood, and vice versa. The probable annual fuel bills for a proposed installation burning either fuel may also be readily calculated when the fuel economies for coal and oil burning plants are determined, and the price of the fuel is known.

To begin with, the power output per unit of fuel will be determined for plants operating under test conditions at full-rated load. This will be followed by similar determinations for fractional and variable loads, taking into account all "stand by" and other losses incidental to actual operating practice.

In considering combined power-plant efficiencies, it is convenient to divide the various kinds of installations into several types, treating all plants falling under the same type in a similar manner. With this idea in view, the following outline is presented:

NON-CONDENSING PLANTS

1. Direct connected.
2. Belted.

SURFACE-CONDENSING PLANTS

1. Direct connected.
    (a) With steam-driven auxiliaries.
    (b) With belted auxiliaries.
    (c) With motor-driven auxiliaries.

2. Belted.
    (a) With steam-driven auxiliaries.
    (b) With belted auxiliaries.
    (c) With motor-driven auxiliaries.

JET-CONDENSING PLANTS

1. Direct connected.
    (a) With steam-driven auxiliaries.
    (b) With belted auxiliaries.
    (c) With motor-driven auxiliaries.

2. Belted.
    (a) With steam-driven auxiliaries.
    (b) With belted auxiliaries.
    (c) With motor-driven auxiliaries.

The only difference in the efficiencies of the direct connected and belted plants lies in the loss of power due to the belt transmission, and this may be allowed for in assuming the efficiency of engine and generator.

The following system of notation will be maintained throughout these calculations:

K = rated capacity of plant in kilowatts.

K_h = Kilowatt hours per barrel of oil.

E_e = mechanical efficiency of engine.

E_g = electrical efficiency of generator.

E_b = efficiency of boilers.

E_c = efficiency of circulating pump.

S_e = steam consumption of main engine in pounds per indicated horse-power hour.

S_f = steam consumption of feed-pumps in pounds per indicated horse-power hour.

S_a = steam consumption of air-pump in pounds per indicated horse-power hour.

S_c = steam consumption of circulating pump in pounds per indicated horse-power hour.

p = total steam consumption of feed-pump in pounds per hour.

P = boiler pressure by gage in pounds.

H = heat value of fuel in B. T. U. per pound.

F = factor of evaporation.

W = pounds circulating water per pound exhaust steam.

h = total head on circulating pump in feet.

NON–CONDENSING PLANTS

WITH the above notation in mind, an expression will be developed first for a non–condensing plant, having steam-driven duplex feed and oil pumps.

The rated capacity of the plant in indicated horse-power will be —

$$\frac{K}{.746\,E_e E_g} \qquad \cdots \quad \cdots \quad \cdots \quad (5)$$

from which the steam consumption of the main engine in pounds per hour will be

$$\frac{K S_e}{.746\,E_e E_g} \qquad \cdots \quad \cdots \quad \cdots \quad (6)$$

Adding the steam used by the feed-pumps per hour, we have

$$\frac{K S_e}{.746\,E_e E_g} + p \;\cdot \qquad \cdots \quad \cdots \quad (7)$$

In addition to the steam consumed by the main engine and feed-pumps, there will be required for the

Oil pumps 1%

Oil burners 3%

Radiation and leakage 3%

    Total 7%

of the entire evaporation, from which the total steam used in the entire plant per hour will be

$$\frac{\dfrac{KS_e}{.746\ E_eE_g} + p}{.93} \qquad (8)$$

Multiplying by the factor of evaporation, we have for the total equivalent evaporation per hour from and at 212° Fahr.

$$F\left(\frac{KS_e}{.746 \times .93\ E_eE_g} + \frac{p}{.93}\right) \qquad (9)$$

Now, HE_b = B. T. U. absorbed by boiler per pound fuel, and 336 HE_b = B. T. U. absorbed by boiler per barrel of fuel (one barrel of oil weighs 336 lbs.), whence the water evaporated into steam in the boiler per barrel oil from and at 212° Fahr. equals

$$\frac{336\ HE_b}{966} \qquad (10)$$

Dividing (9) by (10), we get for the barrels of oil consumed per hour at full-rated load

$$\frac{966\ F\left(\dfrac{KS_e}{.746 \times .93\ E_eE_g} + \dfrac{p}{.93}\right)}{336\ HE_b} \qquad (11)$$

Knowing the capacity of the plant in Kw., and the barrels of oil consumed per hour at full load, the economy of the plant in terms of Kw. hours per barrel of oil is easily obtained by dividing the former by the latter. Dividing K therefore by (11) we have:

$$K_h = \frac{8\ KHE_b}{23\ F\left(\dfrac{KS_e}{.746 \times .93\ E_eE_g} + \dfrac{p}{.93}\right)} \qquad (12)$$

This expression represents correctly the economy of a non-condensing plant, but requires a knowledge of the value of "p" — the steam consumption of the feed-pump per hour. It is possible to express "p" in terms of the other variables in the equation, substituting the expression thus obtained in (12) which can be done as follows:

Assume P = 150 lbs., S_f = 200 lbs., and efficiency of feed-pumps = 80%, then

$$p = \frac{\left(\dfrac{KS_e}{.746\ E_eE_g} + p\right) \times 150 \times 2.31 \times 200}{.93 \times .8 \times 33000 \times 60} \qquad (13)$$

Simplifying,

$$p = \frac{.93\ KS_e + .694\ E_eE_g\ p}{14.752\ E_eE_g} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (14)$$

whence

$$14.058\ E_eE_g\ p = .93\ KS_e \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (15)$$

or

$$p = \frac{.93\ KS_e}{14.058\ E_eE_g} = \frac{.31\ KS_e}{4.686\ E_eE_g} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (16)$$

Substituting (16) in (12) we have kilowatt hours per barrel of oil equal

$$K_k = \frac{8\ KHE_b}{23\ F\left[\left(\dfrac{KS_e}{.694\ E_eE_g}\right) \times \left(\dfrac{.31\ KS_e}{.93 \times 4.686\ E_eE_g}\right)\right]} \quad \cdot \quad \cdot \quad (17)$$

Reducing,

$$K_k = \frac{8\ KHE_b}{23\ F\left(\dfrac{14.058\ KS_eE_eE_g + .694\ KS_eE_eE_g}{.694 \times 14.058\ E_e^2E_g^2}\right)} \quad \cdot \quad \cdot \quad (18)$$

$$K_k = \frac{8\ KHE_b \times .694 \times 14.058\ E_eE_g}{23\ F \times 14.752\ KS_e} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (19)$$

$$K_k = \frac{2.439\ HE_bE_eE_g}{10.603\ FS_e} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (20)$$

$$K_k = \frac{.23\ HE_bE_eE_g}{FS_e} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (21)$$

Equation (21) represents the kilowatt hours per barrel of oil for a non-condensing plant, assuming the boiler pressure to be 150 lbs. by gage and the feed-pumps to require 200 lbs. of steam per indicated horse-power per hour. The fuel consumed by feed-pumps, oil-pumps, oil burners, radiation, leakage, etc., is embodied in the factor .23. Either a belted or direct-connected plant may be solved by this equation, but in case the engine be belted, the efficiency of the belt drive must be included in the efficiency of the engine.

For any Values of "P" and "S_f"

It is possible to develop this formula for any values of "P" and "S_f" as follows:

$$p = \frac{\left[\dfrac{\dfrac{KS_e}{.746\ E_eE_g} + p}{.93}\right] 2.31\ PS_f}{.8 \times 33,000 \times 60} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (22)$$

Simplifying,

$$p = \frac{\left(\dfrac{2.31\ PS_f KS_e}{.746\ E_e E_g}\right) + 2.31\ PpS_f}{.8 \times 33,000 \times 60 \times .93} \qquad \ldots \ldots \quad (23)$$

$$p = \frac{2.31\ PS_f KS_e + .746 \times 2.31\ PpS_f E_e E_g}{.746 \times .8 \times .93 \times 60 \times 33,000\ E_e E_g} \qquad \ldots \ldots \quad (24)$$

hence,

$$.746 \times .8 \times .93 \times 60 \times 33,000\ p\ E_e E_g = 2.31\ PS_f KS_e + .746 \times 2.31$$
$$PpS_f E_e E_g \qquad \ldots \ldots \ldots \quad (25)$$

and

$$p = \frac{2.31\ PS_f KS_e}{.746 \times .8 \times .93 \times 60 \times 33,000\ E_e E_g - .746 \times 2.31\ PS_f E_e E_g} \qquad (26)$$

Reducing,

$$p = \frac{PS_f KS_e}{475,714\ E_e E_g - .746\ PS_f E_e E_g} \qquad \ldots \ldots \quad (27)$$

$$p = \frac{PS_f KS_e}{.746\ E_e E_g\ (637,714 - PS_f)} \qquad \ldots \ldots \quad (28)$$

Substituting (28) in (12) we have kilowatt hours per barrel of oil equal —

$$K_h = \frac{8\ KHE_b}{23\ F\left(\dfrac{KS_e}{.694\ E_e E_g} + \dfrac{PS_f KS_e}{.93 \times .746\ E_e E_g\ (637,714 - PS_f)}\right)} \qquad (29)$$

Reducing and simplifying,

$$K_h = \frac{8\ KHE_b}{23\ F\left(\dfrac{KS_e}{.694\ E_e E_g} + \dfrac{PS_f KS_e}{.93 \times 475,714\ E_e E_g - .93 \times .746\ E_e E_g PS_f}\right)} \qquad (30)$$

$$K_h = \frac{8\ KHE_b}{23\ F\left(\dfrac{637,714\ KS_e - PS_f KS_e + PS_f KS_e}{.93 \times 475,714\ E_e E_g - .93 \times .746\ E_e E_g PS_f}\right)} \qquad (31)$$

$$K_h = \frac{(.93 \times 475,714\ E_e E_g - .93 \times .746\ E_e E_g PS_f)\ 8\ KHE_b}{23\ F \times 637,714\ KS_e} \qquad (32)$$

$$K_h = \frac{.694 \times 8\ HE_b E_e E_g\ (637,714 - PS_f)}{23 \times 637,714\ FS_e} \qquad \ldots \ldots \quad (33)$$

$$K_h = \frac{HE_b E_e E_g\ (637,714 - PS_f)}{2,642,000\ FS_e} \qquad \ldots \ldots \quad (34)$$

$$K_h = \frac{HE_bE_eE_g\left(638 - \dfrac{PS_f}{1000}\right)}{2642\ FS_e} \quad . \quad . \quad . \quad . \quad (35)$$

By substituting for "P" and "S_f" in (35) the values 150 and 200, respectively, it will be found that formula (21) is reproduced, the very slight difference in the constant being due to the manipulation of decimal points and the occasional dropping of a figure in the above development. Equation (35) is correct for any well-designed non-condensing plant, using duplex steam-driven oil and feed-pumps, and oil burners using steam for atomization.

Tables and Graphs for non-Condensing Plants

Figure 21 tabulates results of "$HE_bE_eE_g$" for various values of each variable. Figure 22 sets forth calculations for "FS_e" which may be used profitably in applying equation (21). Figure 23 is a graphical representation of equation (21). Having selected proper values for "$HE_bE_eE_g$" from Fig. 21, and "FS_e" from Fig. 22, the economy of the plant may be read directly from Fig. 23. Figures 21 and 22 may also be used in connection with equation (35). In solving this equation with the accompanying graphs, first select the proper value of "$HE_bE_eE_g$" from Fig. 21, knowing which, the numerator of the fraction may be read directly from Fig. 24 for any value of "PS_f." Referring then to Fig. 25, and having obtained from Fig. 22 the correct result for "FS_e," the economy of the plant will be found plotted upon the left-hand side of the sheet.

It should be remembered that these equations apply only to test conditions at full-rated load, and that a margin of safety should always be applied after having obtained the final result for an assumed set of conditions. This margin should never be less than 5% and may be as high as 10 or 15%, depending upon the degree of conservatism in the selection of the various variables assumed.

Illustration

To illustrate the method in which the above tables are used, the following illustration is given. Required — the economy of a non-condensing steam-power plant, under the following conditions:

$$E_eE_g = 90\%.$$
$$E_b = 80\%.$$
$$S_e = 26\ \text{lbs.}$$
$$P = 140\ \text{lbs.}$$
$$H = 18,850\ \text{B. T. U.}$$
$$S_f = 150\ \text{lbs.}$$

From Fig. 21, we have —

$$HE_bE_eE_g = 13,572.$$

FIG. 21. — VALUES OF "$HE_b E_e E_g$"

Boiler Efficiency "E_b" — %	Combined Efficiency of Engine and Generator ($E_e E_g$) — %													
	80%	81	82	83	84	85	86	87	88	89	90	91	92	93

Calorific Value of Oil per Pound = "H" = 18000 B.T.U.

	80%	81	82	83	84	85	86	87	88	89	90	91	92	93
65	9360	9477	9594	9711	9828	9945	10062	10179	10296	10413	10530	10647	10764	10881
66	9504	9623	9742	9860	9979	10098	10217	10336	10454	10573	10692	10811	10930	11048
67	9648	9769	9889	10010	10130	10251	10372	10492	10613	10733	10854	10975	11095	11216
68	9792	9914	10037	10159	10282	10404	10526	10649	10771	10894	11016	11138	11261	11383
69	9936	10060	10184	10309	10433	10557	10681	10805	10930	11054	11178	11302	11426	11551
70	10080	10206	10332	10458	10584	10710	10836	10962	11088	11214	11340	11466	11592	11718
71	10224	10352	10480	10607	10735	10863	10991	11119	11246	11374	11502	11630	11758	11885
72	10368	10498	10627	10757	10886	11016	11146	11275	11405	11534	11664	11794	11923	12053
73	10512	10643	10775	10906	11038	11169	11300	11432	11563	11695	11826	11957	12089	12220
74	10656	10789	10922	11056	11189	11322	11455	11588	11722	11855	11988	12121	12254	12388
75	10800	10935	11070	11205	11340	11475	11610	11745	11880	12015	12150	12285	12420	12555
76	10944	11081	11218	11354	11491	11628	11765	11902	12038	12175	12312	12449	12586	12722
77	11088	11227	11365	11504	11642	11781	11920	12058	12197	12335	12474	12613	12751	12890
78	11232	11372	11513	11653	11794	11934	12074	12215	12355	12496	12636	12776	12917	13057
79	11376	11518	11660	11803	11945	12087	12229	12371	12514	12656	12798	12940	13082	13225
80	11520	11664	11808	11952	12096	12240	12384	12528	12672	12816	12960	13104	13248	13392
81	11664	11810	11956	12101	12247	12393	12539	12685	12830	12976	13122	13268	13414	13559
82	11808	11956	12103	12251	12398	12546	12694	12841	12989	13136	13284	13432	13579	13727
83	11952	12101	12251	12400	12550	12699	12848	12998	13147	13297	13446	13595	13745	13894

"H" = 18500 B.T.U.

	80%	81	82	83	84	85	86	87	88	89	90	91	92	93
65	9620	9740	9861	9981	10101	10221	10342	10462	10582	10702	10823	10943	11063	11183
66	9768	9890	10012	10134	10256	10379	10501	10623	10745	10867	10989	11111	11233	11355
67	9916	10040	10164	10288	10412	10536	10660	10784	10908	11032	11156	11279	11403	11527
68	10064	10190	10316	10441	10567	10693	10819	10945	11070	11196	11322	11448	11574	11699
69	10212	10340	10467	10595	10723	10850	10978	11106	11233	11361	11489	11616	11744	11871
70	10360	10490	10619	10749	10878	11008	11137	11267	11396	11526	11655	11785	11914	12044
71	10508	10640	10771	10902	11033	11165	11296	11427	11559	11690	11822	11953	12084	12216
72	10656	10790	10922	11056	11189	11322	11455	11588	11722	11855	11988	12121	12254	12388
73	10804	10940	11074	11209	11344	11479	11614	11749	11884	12019	12155	12290	12425	12560
74	10952	11089	11226	11363	11500	11637	11773	11910	12047	12184	12321	12458	12595	12732
75	11100	11239	11378	11516	11655	11794	11933	12071	12210	12349	12488	12626	12765	12904
76	11248	11389	11529	11670	11810	11951	12092	12232	12373	12513	12654	12795	12935	13076
77	11396	11538	11681	11823	11966	12108	12251	12393	12536	12678	12821	12963	13105	13248
78	11544	11688	11833	11977	12121	12266	12410	12554	12698	12843	12987	13131	13276	13420
79	11692	11838	11984	12130	12277	12423	12569	12715	12861	13007	13154	13300	13446	13592
80	11840	11988	12136	12284	12432	12580	12728	12876	13024	13172	13320	13468	13616	13764
81	11988	12138	12288	12438	12587	12737	12887	13037	13187	13337	13487	13636	13786	13936
82	12136	11288	12439	12591	12743	12895	13046	13198	13350	13501	13653	13805	13956	14108
83	12284	12438	12591	12745	12898	13052	13205	13359	13512	13666	13820	13973	14127	14280

"H" = 18850 B.T.U.

	80%	81	82	83	84	85	86	87	88	89	90	91	92	93
65	9802	9925	10047	10170	10292	10415	10537	10660	10782	10904	11027	11150	11272	11395
66	9953	10077	10202	10326	10450	10575	10699	10824	10948	11073	11197	11321	11446	11570
67	10104	10230	10356	10483	10609	10735	10861	10988	11114	11240	11367	11493	11619	11745
68	10254	10383	10511	10639	10767	10895	11023	11152	11280	11408	11536	11664	11793	11921
69	10405	10535	10665	10795	10926	11056	11186	11316	11446	11576	11706	11836	11966	12096
70	10556	10688	10819	10952	11084	11216	11348	11480	11612	11744	11876	12008	12139	12271
71	10707	10841	10975	11108	11242	11376	11510	11644	11777	11911	12045	12179	12313	12447
72	10858	10993	11129	11265	11401	11536	11672	11808	11943	12079	12215	12351	12486	12622
73	11008	11146	11284	11421	11559	11696	11834	11972	12109	12247	12385	12522	12660	12797
74	11159	11299	11438	11578	11717	11857	11996	12136	12275	12415	12554	12694	12833	12973
75	11310	11451	11593	11734	11876	12017	12158	12300	12441	12582	12724	12865	13007	13148
76	11461	11604	11747	11891	12034	12177	12320	12464	12607	12750	12893	13037	13180	13323
77	11612	11757	11902	12047	12192	12337	12483	12628	12773	12918	13063	13208	13353	13498
78	11762	11909	12057	12204	12351	12498	12645	12792	12939	13086	13233	13380	13527	13674
79	11913	12062	12211	12360	12509	12658	12807	12956	13105	13253	13402	13551	13700	13849
80	12064	12215	12365	12515	12666	12817	12968	13119	13269	13420	13572	13722	13873	14023
81	12215	12367	12520	12673	12826	12978	13131	13284	13436	13589	13742	13894	14047	14200
82	12366	12520	12675	12829	12984	13138	13293	13448	13602	13757	13911	14066	14220	14375
83	12516	12673	12829	12986	13142	13299	13455	13612	13768	13924	14081	14237	14394	14550

The boiler pressure equals 140 lbs., and inasmuch as the exhaust steam from the main engine will be available for heating the feed-water, the temperature of the latter should not be less than 212° Fahr. It is not safe, however, to assume such a high temperature, on account of leakage, radiation, etc. Say, therefore, that 200° Fahr. will be the final feed-water temperature. Under these conditions, the factor of evaporation "F"
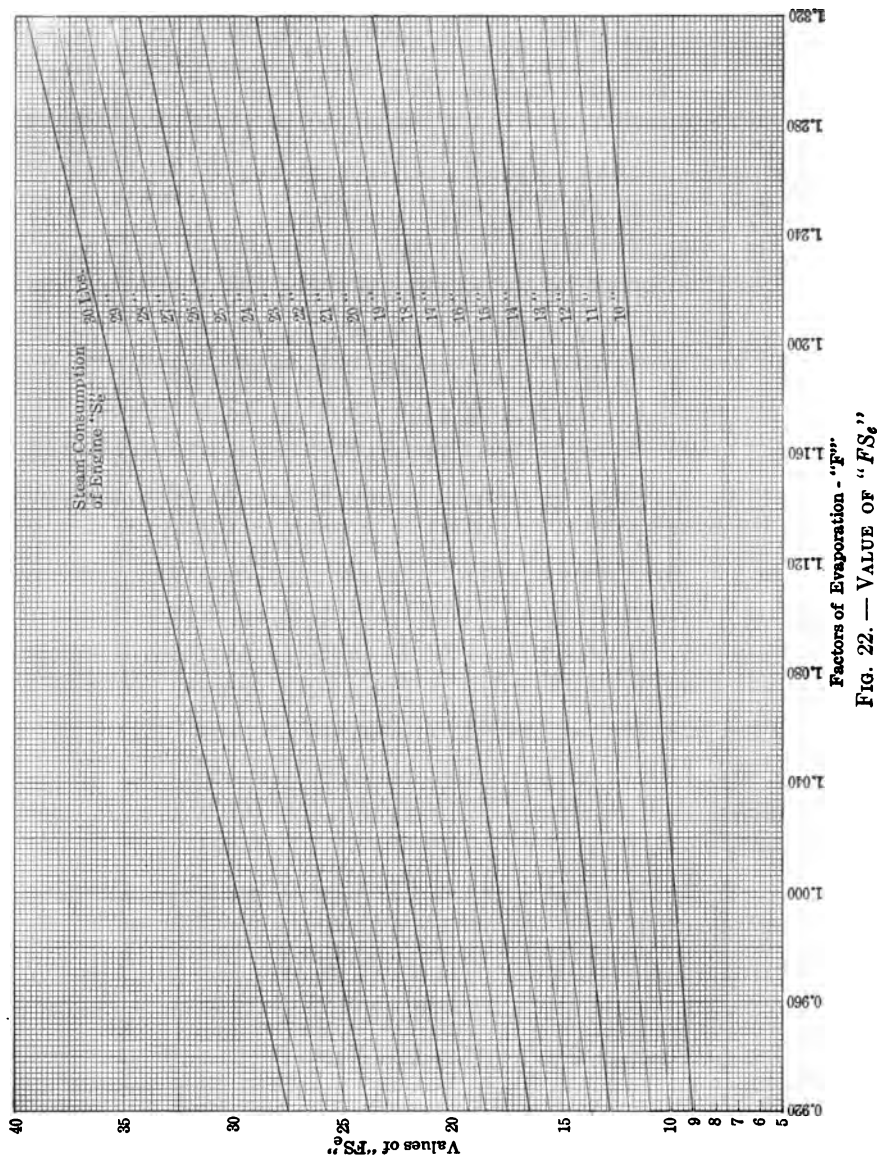
Fig. 22. — Value of "FS_e"

may be selected from any of the standard tables, and will be found to be 1.06. We also have

$$PS_f = 21,000.$$

Referring now to Fig. 24, we have (when $HE_bE_eE_g = 13,572$ and $PS_f = 21,000$) the numerator of equation (35). This is found to be for
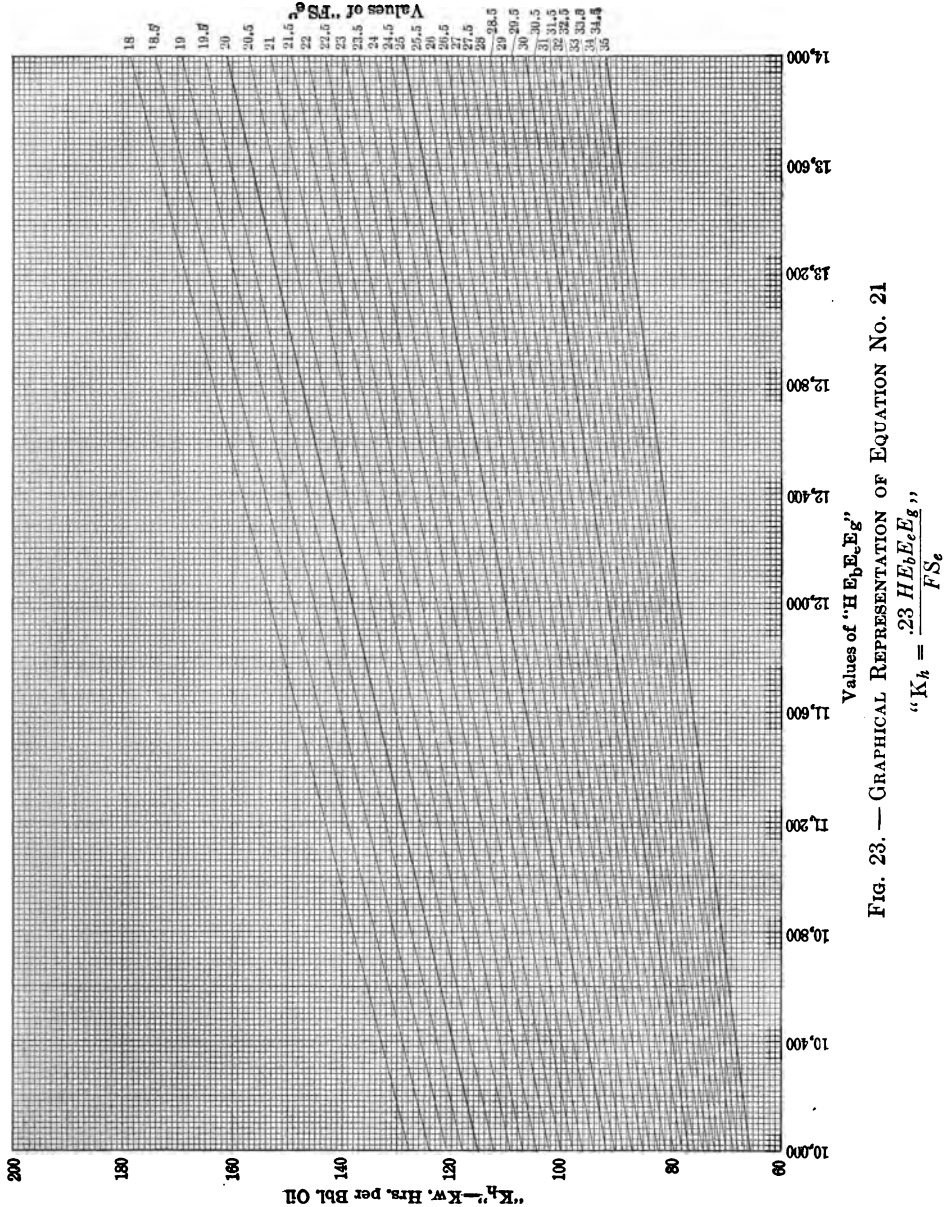
FIG. 23. — GRAPHICAL REPRESENTATION OF EQUATION NO. 21

Values of "$HE_bE_eE_g$"

$$"K_h = \frac{.23\ HE_bE_eE_g}{FS_e}"$$

the present conditions about 8,374,000.   From Fig. 22 we now have $FS_e =$ 27.56, which gives us sufficient data to obtain from Fig. 25 the final result of 115 Kw. hours per barrel of oil.

Fig. 24. — Graphical Representation of Numerator $$\text{``}HE_bE_eE_g\left(638-\frac{PS_f}{1000}\right)\text{''}$$
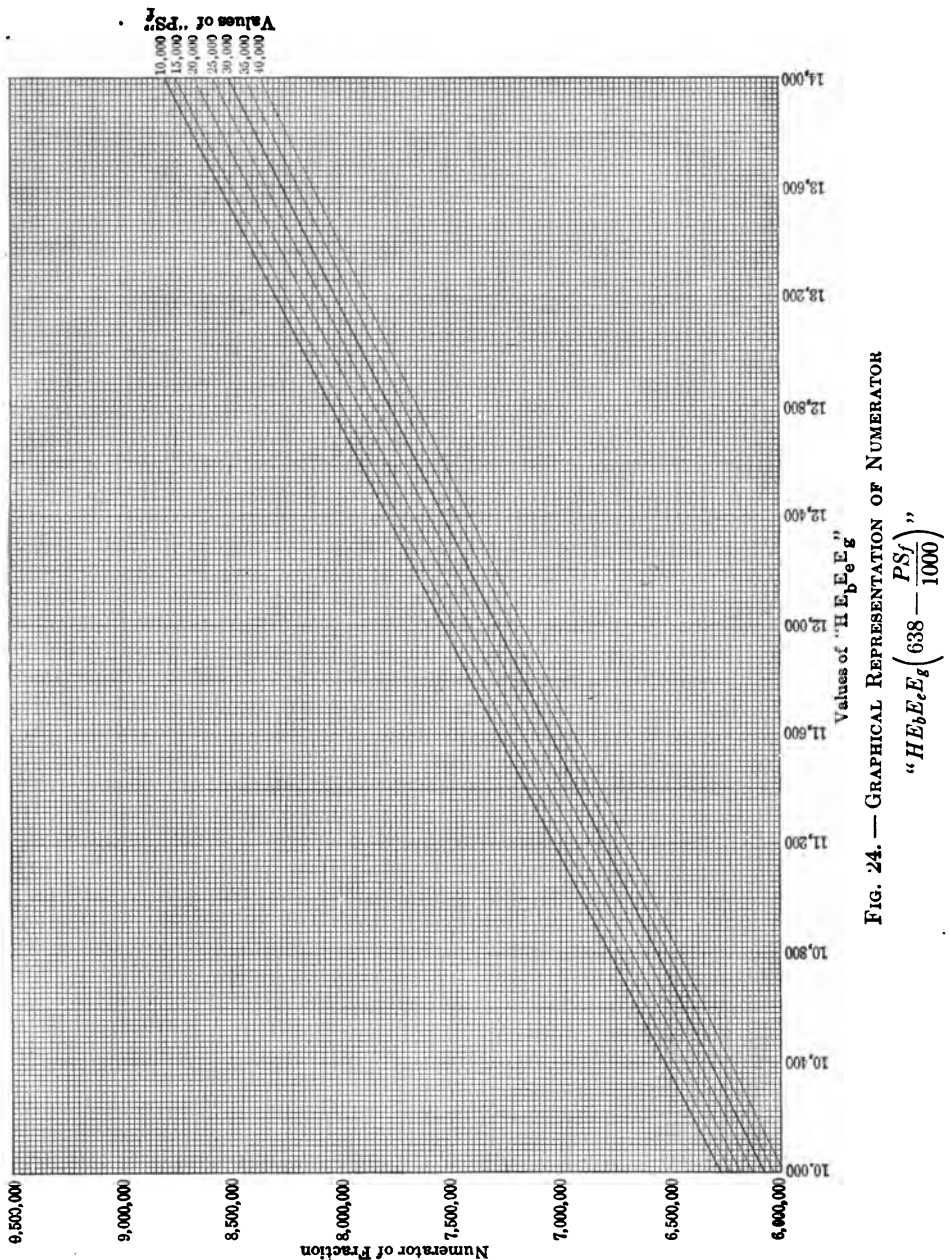
In the above example, the values for the different factors entering into the net plant economy have been directly assumed, but in actual practice it is seldom, if ever, that this data is directly obtainable. The size of the plant is given with a general description of the apparatus included, and

Fig. 25. — Graphical Representation of Equation No. 35. (Non-Condensing Plants)

$$K_h = \frac{HE_b E_e E_g \left(638 - \dfrac{PS_f}{1000}\right)}{2642 \, FS_e}$$
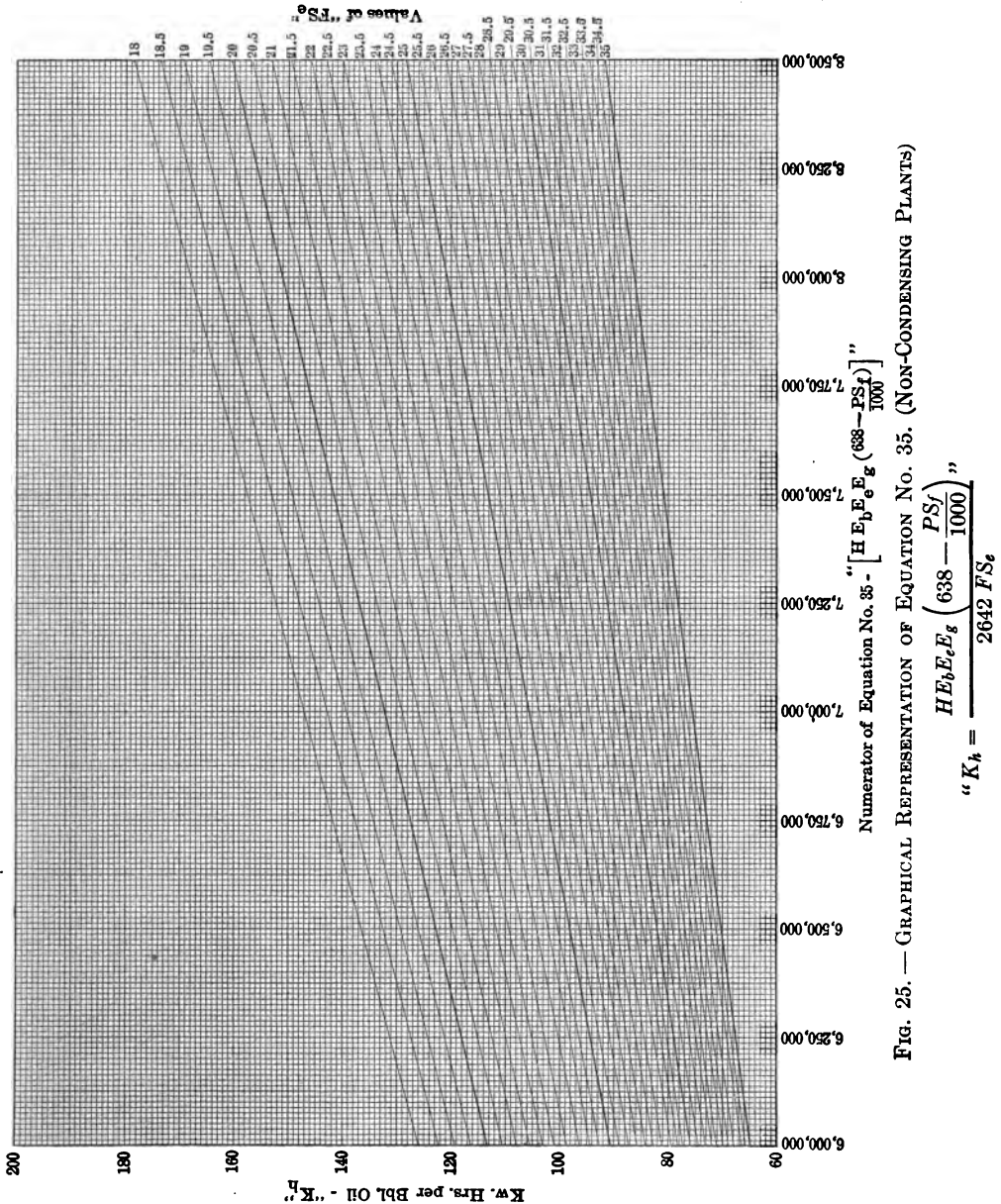
this description must serve as a basis for the necessary calculations. A knowledge of the economical capabilities of each type of auxiliary machine, as well as of the main units, is therefore essential to a proper solution of the power-plant economy problem, and it is for this reason that the performance of each type of machine was taken up at some length in the preceding pages.

The above equations, tables and graphs are sufficient for all non-condensing plants using steam-driven feed and oil-pumps, and inasmuch as it is considered poor practice to drive these auxiliaries in any other way, due to regulation losses and difficulties, equations have not been presented for power-driven pumps. It should be remembered, however, that these figures apply only to modern plants having well-designed piping with adequately covered joints, valves, and fittings. In case a belted plant is used, it is only necessary to include the belt loss with the combined efficiency of engine and generator. This loss varies between about 4% and 8%, depending upon conditions.

Following, condensing plants will be investigated in general accordance with the method outlined above.

SURFACE–CONDENSING PLANTS

When condensing apparatus is added to a plant, certain additional auxiliaries are required, as already described. If the plant be surface condensing, an air and circulating pump are necessary, and if jet condensing, one pump only suffices, provided the required pressure is available for the injection water. Regarded in this light, there is a certain addition to the steam and fuel consumption of a non-condensing plant when condensers are added, of which the economy of the condenser auxiliaries is a measure. The real economic gain is, of course, due to the greatly decreased value of "S_e" when the engine is operated under condensing conditions. The condensing plant calculations will be based on this consideration, attention being first directed to surface-condensing plants, using steam-driven auxiliaries.

Surface-Condensing Plants — Steam-Driven Air and Circulating Pumps

The power required for driving the air and circulating pumps of a surface-condensing plant and the steam consumption resulting therefrom were fully reviewed under their proper headings. Based on the data therein given, the following calculations will be easily followed:

The I. H. P. developed in the steam cylinder of the air-pump equals,

$$\frac{.00024\, KS_e}{.6 \times .746\, E_e E_g} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (36)$$

from which the steam consumption of the air-pump in pounds per hour is,

$$\frac{.00024 \ KS_eS_a}{.6 \times .746 \ E_eE_g} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (37)$$

The I. H. P. developed in the steam cylinder of the circulating pump equals,

$$\frac{WhKS_e}{.746 \ E_eE_gE_c \times 60 \times 33,000} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (38)$$

which gives the steam consumption in pounds per hour of the circulating pump as follows:

$$\frac{WhKS_eS_c}{33,000 \times 60 \times .746 \ E_eE_gE_c} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (39)$$

Adding (37) and (39) we have the steam required by both air and circulating pumps per hour to be,

$$\frac{.0004KS_eS_a}{.746E_eE_g} + \frac{WhKS_eS_c}{.746 \times 1,980,000 \ E_eE_gE_c} \quad \cdot \quad \cdot \quad \cdot \quad (40)$$

or

$$\frac{792 \ EKS_eS_a + WhKS_eS_c}{1,477,080 \ E_eE_gE_c} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (41)$$

The feed-pump is the next auxiliary to be considered. This is figured in exactly the same manner as in non-condensing installations. It is first necessary, however, to add together the steam consumption of the main engine, the air-pump, and the circulating pump. This addition is represented by the sum of equations (6) and (41) as follows:

$$\frac{KS_e}{.746 \ E_eE_g} + \frac{792 \ E_cKS_eS_a + WhKS_eS_c}{1,477,080 \ E_eE_gE_c} \quad \cdot \quad \cdot \quad \cdot \quad (42)$$

which, simplified, is represented by,

$$\frac{KS_e \ (1,980,000 \ E_c + 792 \ E_cS_a + WhS_c)}{1,477,080 \ E_eE_gE_c} \quad \cdot \quad \cdot \quad \cdot \quad (43)$$

In estimating the steam consumption of the feed-pump and of the entire plant, the same assumptions relative to the steam required by the oil-pumps and burners, radiation and leakage, etc., will be maintained as in the non-condensing plants. That is, 7% of the total steam generated is used in this manner. From these considerations, it will be seen that the total steam used per hour by the feed-pump, equals,

$$p = \frac{\left[\dfrac{KS_e\,(1{,}980{,}000\,E_c + 792\,E_cS_a + WhS_c)}{.93 \times 1{,}477{,}080\,E_eE_gE_c} + \dfrac{p}{.93}\right]2.31\,PS_f}{.8 \times 33{,}000 \times 60} \quad . \quad (44)$$

Simplifying and reducing,

$$p = \frac{2.31\,PS_fKS_e\,(1{,}980{,}000\,E_c + 792\,E_cS_a + WhS_c)}{1{,}477{,}080\,E_eE_gE_c\,(1{,}473{,}120 - 2.31\,PS_f)} \quad . \quad (45)$$

By dividing the sum of equations (43) and (45) by .93, and multiplying by "F" the total water evaporated in the complete plant per hour from and at 212° Fahr., will result. This will equal,

$$\frac{FKS_e\,[(1{,}980{,}000\,E_c + 792\,E_cS_a + WhS_c)\,(1{,}473{,}120 - 2.31\,PS_f) + 2.31\,PS_f\,(1{,}980{,}000\,E_c + 792\,E_cS_a + WhS_c)]}{.93 \times 1{,}477{,}080\,E_eE_gE_c\,(1{,}473{,}120 - 2.31\,PS_f)} \quad (46)$$

Dividing (46) by (10) the barrels of fuel oil consumed by the entire plant per hour when operating at full-rated load equals,

$$\frac{23\,FKS_e\,[(1{,}980{,}000\,E_c + 792\,E_cS_a + WhS_c)\,(1{,}473{,}120 - 2.31\,PS_f) + 2.31\,PS_f\,(1{,}980{,}000\,E_c + 792\,E_cS_a + WhS_c)]}{8 \times .93 \times 1{,}477{,}080\,HE_bE_eE_gE_c\,(1{,}473{,}120 - 2.31\,PS_f)} \quad (47)$$

whence the economy of the complete plant in terms of Kw. hours per barrel of oil is found to be

$$K_h = \frac{.93 \times 8 \times 1{,}477{,}080\,HE_bE_eE_gE_c\,(1{,}473{,}120 - 2.31\,PS_f)}{23\,FS_e\,[(1{,}980{,}000\,E_c + 792\,E_cS_a + WhS_c)\,(1{,}473{,}120 - 2.31\,PS_f) + 2.31\,PS_f\,(1{,}980{,}000\,E_c + 792\,E_cS_a + WhS_c)]} \quad (48)$$

Simplifying,

$$K_h = \frac{.93 \times 8 \times 1{,}477{,}080\,HE_bE_eE_g}{23\,FS_e\left[\left(1{,}980{,}000 + 792\,S_a + \dfrac{WhS_c}{E_c}\right) + \left(1{,}980{,}000 + 792\,S_a + \dfrac{WhS_c}{E_c}\right) \times \left(\dfrac{2.31\,PS_f}{1{,}473{,}120 - 2.31\,PS_f}\right)\right]} \quad . \quad . \quad . \quad . \quad (49)$$

or,

$$K_h = \frac{10{,}989{,}475.2\,HE_bE_eE_g}{792 \times 23\,FS_e\left[\left(\dfrac{2.31\,PS_f}{1{,}473{,}120 - 2.31\,PS_f} + 1\right)\left(2500 + S + \dfrac{WhS_c}{792\,E_c}\right)\right]} \quad (50)$$

Reducing,

$$K_h = \frac{603\,HE_bE_eE_g}{FS_e\left(2500 + S_a + \dfrac{WhS_c}{E_c}\right)\left(\dfrac{1}{1 - .00000157\,PS_f}\right)} \quad . \quad (51)$$

which gives

$$K_h = \frac{603\ HE_bE_cE_g\ (1 - .00000157\ PS_f)}{FS_e\left(2500 + S_a + \dfrac{WhS_c}{792\ E_c}\right)} \quad . \quad . \quad . \quad . \quad (52)$$

Reducing equation (52) to the same form as that for non-condensing plants, we get, kilowatt hours per barrel of oil —

$$K_h = \frac{HE_bE_cE_g\left(638 - \dfrac{PS_f}{1000}\right)}{1.06\ FS_e\left(2500 + S_a + \dfrac{WhS_c}{792\ E_c}\right)} \quad . \quad . \quad . \quad (53)$$

This formula will solve any condensing plant having steam-driven auxiliaries. It will be noted that while the efficiency of the circulating pump is denoted by "E_c," there is nothing to represent the efficiencies of the air or feed-pumps. The latter are, however, included in the different constants. The feed-pump efficiency, it will be remembered, was assumed to be 80% throughout, and that of the air pump 60%. These efficiencies appear to be very nearly correct for any types of steam-driven pumps, but it will be found that a slight variation from these values will affect the final result, only by an inappreciable amount. The efficiency of the circulating pumps, however, varies considerably, depending upon the type of unit installed. While a direct acting arrangement may operate with an efficiency of 80%, a direct connected engine-driven centrifugal outfit would have a combined engine and pump efficiency of only from 40 to 60%, depending upon the size and speed of the unit and the head under which it pumped. For this reason it was necessary to leave the circulating pump efficiency to separate determinations in each special case.

TABLES AND GRAPHS

In applying equation (53) the accompanying tables and graphs will be found useful. The numerator of the fractions may be obtained from Figs. 21 and 24, in an exactly similar manner as for non-condensing plants. "FS_e" may be obtained from Fig. 22, and from Fig. 26, knowing which, the resultant economy of the plant may be read from Fig. 27. In using this graph, first locate the proper value of $\dfrac{``WhS_c"}{792\ E_c}$ upon the lower edge of the sheet, then move vertically until one of the parallel lines is met, denoting the correct value of S_a, thence horizontally to one of the straight diverging lines, representing "FS_e," from which, by moving again vertically to the intersection of one of the curved lines designating the proper value for the numerator of the fraction, the final result in terms of Kw.

Fig. 26. — Values of $\dfrac{\text{"}WhS_c\text{"}}{792\ E_c}$

(For Surface Condensing Plants — Steam Driven Auxiliaries)

"S_c"	Values of "Wh"						
	500	700	1000	1500	2000	2500	3000
"E_c" = 40%							
30	47.35	66.29	94.70	142.05	189.39	236.74	284.09
40	63.13	88.38	126.26	189.39	252.53	315.66	378.79
50	78.92	110.48	157.83	236.74	315.66	394.58	473.48
60	94.70	132.59	189.39	284.09	378.79	473.48	568.18
100	157.83	220.96	315.66	473.48	631.31	789.15	946.97
150	236.74	331.44	473.48	710.23	946.97	1183.70	1420.45
"E_c" = 45%							
30	42.09	58.92	84.17	126.26	168.35	210.44	252.53
40	56.12	78.56	112.23	168.35	224.47	280.59	336.70
50	70.15	98.20	140.29	210.44	280.59	350.74	420.88
60	84.18	117.86	168.35	252.53	336.70	420.88	505.05
100	140.29	196.41	280.59	420.88	561.16	701.45	841.75
150	210.44	294.62	420.88	631.31	841.75	1052.20	1262.63
"E_c" = 50%							
30	37.88	53.03	75.76	113.64	151.52	189.39	227.27
40	50.50	70.70	101.01	151.52	202.02	252.53	303.03
50	63.13	88.38	126.26	189.39	252.53	315.67	378.79
60	75.76	106.06	151.52	227.27	303.03	378.79	454.55
100	126.26	176.77	252.53	378.79	505.05	631.31	757.58
150	189.39	265.15	378.79	568.18	757.58	946.96	1136.36
"E_c" = 55%							
30	34.44	48.21	68.87	103.31	137.74	172.18	206.61
40	45.91	64.28	91.83	137.74	183.65	229.57	275.48
50	57.39	80.35	114.78	172.18	229.57	286.96	344.35
60	68.87	96.42	137.74	206.61	275.48	344.35	413.22
100	114.78	160.70	229.57	344.35	459.14	573.92	688.71
150	172.18	241.05	344.35	516.53	688.71	860.88	1033.06
"E_c" = 60%							
30	31.57	44.20	63.13	94.70	126.26	157.83	189.39
40	42.09	58.93	84.18	126.26	168.36	210.44	252.53
50	52.61	73.65	105.22	157.83	210.44	263.05	315.66
60	63.13	88.38	126.26	189.39	252.53	315.66	378.79
100	105.22	147.31	210.44	315.66	420.88	526.10	631.31
150	157.83	220.96	315.66	473.48	631.31	789.14	946.97
"E_c" = 80%							
30	23.68	33.15	47.35	71.03	94.70	118.37	142.05
40	31.57	44.20	63.13	94.70	126.26	157.83	189.39
50	39.46	55.24	78.92	118.37	157.83	197.29	236.74
60	47.35	66.29	94.70	142.05	189.39	236.74	284.09
100	78.92	110.48	157.83	236.74	315.66	394.58	473.48
150	118.37	165.72	236.74	355.12	473.48	591.85	710.23

hours per barrel of oil will be read upon the left-hand side of the figure. In case it is desired to find, for any reason, a certain part of the fraction before reaching the final result, the expression

$$25 \times S_a + \frac{WhS_c}{792\,E_c}$$

will be found plotted vertically upon the right hand, and the denominator of the fractions upon the upper side of the graph.

Suppose, in a given installation, for example, sufficient data be at hand to determine from the tables the following values:

Numerator of
equation (53) . . . . , ... 10,000,000
FS_e . . . . 18.
S_a . . . . 40.
$\dfrac{WhS_c}{792E_c}$ . . . . 50.

Then referring to Fig. 27 and moving vertically along the line which designates $\dfrac{Wh\,S_c}{792\,E_c}$ as 50 until the parallel line marked $S_a = 40$ is reached, the expression "$2,500 + S_a + \dfrac{WhS_c}{792\,E_c}$" may be read, if desired, upon the right-hand side, and will be found to be 2590.

From this point of intersection follow horizontally until the line denoting FS_e is equal to 18 is reached, when the denominator of the fraction may be read upon the upper edge of the figure. For the present conditions, this will equal about 49,420. From the intersection last found, proceed vertically until meeting the curved line marked 10,000,000, when the final result may be read directly opposite on the left-hand side of the sheet as nearly 203 Kw. hours per barrel of oil.

As in the case of non-condensing plants, it should be remembered that these figures are theoretical and require protection by a safe margin. If all the losses and efficiencies are exactly the same as assumed for the entire test, the result will obviously be exactly as shown by the formula, but due to possible slight fluctuations in the load, and the impossibility of maintaining each and every piece of apparatus at a point of maximum efficiency during the entire test, certain discrepancies are likely to occur, and these should be guarded against by applying a certain margin of safety to the final results.

SURFACE-CONDENSING PLANTS — POWER-DRIVEN AUXILIARIES

Attention will now be directed to surface-condensing plants having power-driven auxiliaries. By "power-driven auxiliaries" is meant auxiliaries either belted to the main engine shaft or direct connected or geared

to motors.   It was above suggested that even when the air and circula-
ting pumps were motor driven, it was considered the better practice to
maintain steam-driven feed and oil-pumps, and the following calculations
will be based upon this condition.

With motor-driven air and circulating pumps, and with the character
of plant above outlined, steam will be consumed —

        (1) By the main engine.
        (2) By the feed-pumps.
        (3) By the oil-pumps.
        (4) By the oil burners.
        (5) By radiation and leakage.

The last three items will be assumed at 7% as for the other plants.
The first item will comprise, not only the necessary steam for the pro-
duction of sufficient power to operate the generator at full load, but in
addition, that necessary for the power required, by the air and circulating
pumps.   This latter quantity will involve the efficiencies of the main
engine and generator, the motors and auxiliaries, and the steam consump-
tion of the main engine.

The following additional notation will be used in the calculations relat-
ing to power-driven auxiliaries:

E_{cm} = Efficiency of circulating pump motor.

E_{am} = Efficiency of air-pump motor.

x = Total steam in pounds per hour required .by the main engine for
      operating the air-pumps.

y = Total steam in pounds per hour required by the main engine for
      operating the circulating pump.

It should be constantly borne in mind that in all cases where steam-
driven auxiliaries are used, "E_c" represents the net efficiency of the com-
plete circulating pump unit regardless of whether or not it be separately
driven by a steam engine or whether direct acting.   For certain reasons,
however, "E_c" will only represent the efficiency of the pump itself, not
including the motor when this auxiliary is power driven.   For instance,
if the circulating pump be centrifugal, direct connected to a motor, "E_c"
will represent the efficiency of the circulating pump, and "E_{cm}" the
efficiency of the motor, while if the same pump be direct connected to an
engine, as would be the case in a plant having steam-driven auxiliaries,
"E_c" would represent the combined efficiency of pump and engine.

From the above considerations, it will be seen that the total quantity
of steam used per hour by the main engine, the air-pump, and circulating
pump would be —

$$\frac{KS_e}{.746\ E_e E_g} + x + y \quad . \quad . \quad . \quad . \quad . \quad (54)$$

The actual pump horse-power of the air pump is

$$\frac{.00024 \, KS_e}{.746 \, E_e E_g} \qquad \cdots \cdots \cdots \cdots \quad (55)$$

or the I. H. P. in the main engine cylinder due to the air-pump,

$$\frac{.00024 \, KS_e}{.6 \times .746 \, E_e^2 E_g^2 E_{am}} \qquad \cdots \cdots \cdots \cdots \quad (56)$$

The actual pump horse-power of the circulating pump equals

$$\frac{KS_e Wh}{.746 \, E_e E_g \times 60 \times 33{,}000} \qquad \cdots \cdots \cdots \quad (57)$$

from which the I. H. P. in the main engine, due to the circulating pump, will be

$$\frac{KS_e Wh}{1{,}980{,}000 \times .746 \, E_e^2 E_g^2 E_c E_{cm}} \qquad \cdots \cdots \quad (58)$$

from (56)

$$x = \frac{.0004 \, KS_e^2}{.746 \, E_e^2 E_g^2 E_c E_{cm}} \qquad \cdots \cdots \cdots \quad (59)$$

and from (58)

$$y = \frac{KS_e^2 Wh}{1{,}980{,}000 \times .746 \, E_e^2 E_g^2 E_c E_{cm}} \qquad \cdots \cdots \quad (60)$$

Substituting (59) and (60) in (54) and simplifying, we have the steam used per hour by main engine, air-pump, and circulating pump, equals —

$$\frac{KS_e(1{,}980{,}000 \, E_e E_g E_c E_{am} E_{cm} + 1{,}980{,}000 \times .0004 \, E_c E_{cm} S_e + WhS_e E_{am})}{1{,}980{,}000 \times .746 \, E_e^2 E_g^2 E_c E_{am} E_{cm}} \quad (61)$$

The steam required by the feed-pumps is obtained in the same manner as that for steam-driven auxiliaries, thus —

$$p = \frac{\left[\dfrac{KS_e(1{,}980{,}000 \, E_e E_g E_c E_{am} E_{cm} + 792 \, S_e E_c E_{cm} + S_e E_{am} Wh)}{.93 \times 1{,}980{,}000 \times .746 \, E_e^2 E_g^2 E_c E_{am} E_{cm}} + \dfrac{p}{.93}\right] 2.31 PS_f}{.8 \times 60 \times 33{,}000}$$

whence $\qquad (62)$

$$p\left(\frac{.8 \times 1{,}980{,}000 - 2.31 \, PS_f}{.93 \times .8 \times 1{,}980{,}000}\right) =$$

$$\frac{2.31 \, PS_f KS_e (1{,}980{,}000 \, E_e E_g E_c E_{am} E_{cm} + 792 \, S_e E_c E_{cm} + S_e E_{am} Wh)}{.8 \times .746 \times .93 \times 1{,}980{,}000^2 E_e^2 E_g^2 E_c E_{am} E_{cm}} \quad (63)$$

whence,

$$p = \frac{2.31 \, PS_f KS_e (1{,}980{,}000 \, E_e E_g E_c E_{am} E_{cm} + 792 \, S_e E_c E_{cm} + S_e E_{am} Wh)}{1{,}477{,}080 \, E_e^2 E_g^2 E_c E_{am} E_{cm} (1{,}473{,}120 - 2.31 \, PS_f)} \quad (64)$$

Adding (61) and (64), dividing by .93, multiplying by "F" and simplifying, gives us the total steam required by the entire plant per hour from and at 212° Fahr., as follows:

$$\frac{FKS_e\left[1,473,120\left(1,980,000\,E_eE_gE_cE_{am}E_{cm}+792\,S_eE_cE_{cm}+S_eE_{am}Wh\right)\right]}{.93\times1,477,080\,E_e^2E_g^2E_cE_{am}E_{cm}\left(1,473,120-2.31\,PS_f\right)} \quad (65)$$

From the above, kilowatt hours per barrel of oil can be obtained by dividing by $\dfrac{8\,HE_b}{23}$ and again dividing "K" by the quotient, thus: Kilowatt hours per barrel of oil equals,

$$K_h=\frac{.93\times8\times1,477,080\,HE_bE_e^2E_g^2E_cE_{am}E_{cm}\left(1,473,120-2.31\,PS_f\right)}{792\times1,473,120\times23\,FS_e\left(2500\,E_eE_gE_cE_{am}E_{cm}+S_eE_cE_{cm}+\dfrac{S_eE_{am}Wh}{792}\right)} \quad (66)$$

Simplifying,

$$K_h=\frac{HE_bE_e^2E_g^2\left(1,473,120-2.31\,PS_f\right)}{792\times3.083\,FS_e\left(2500\,E_eE_g+\dfrac{S_e}{E_{am}}+\dfrac{WhS_e}{792\,E_cE_{cm}}\right)} \quad . \quad (67)$$

Reducing,

$$K_h=\frac{HE_bE_eE_g\left(638-\dfrac{PS_f}{1000}\right)}{1.06\,FS_e\left(2500+\dfrac{S_e}{E_eE_gE_{am}}+\dfrac{WhS_e}{792\,E_eE_gE_cE_{cm}}\right)} \quad . \quad (68)$$

Equation (68) is correct for any surface-condensing plant having motor-driven auxiliaries. Should the auxiliaries be belted, the combined efficiencies of the main generator and auxiliary motors would be eliminated so that wherever the expressions "E_gE_{am}" and "E_gE_{cm}" occur, the efficiencies of the air-pump belt drive, and circulating pump belt drive, respectively, will be substituted.

An interesting comparison may be made between equations (53) and (68) — both representing condensing plants, the former with steam-driven and the latter with electrically-driven auxiliaries. It will be noted that the steam consumption of the air and circulating pumps in (53), viz.: "S_a" and "S_e" are replaced in (68) by the expressions $\dfrac{S_e}{E_eE_gE_{am}}$ and $\dfrac{S_e}{E_eE_gE_{cm}}$ and a little consideration will show that this substitution is exactly what would be expected. It is clear that in a plant of any size, $\dfrac{S_e}{E_eE_gE_{am}}$ would be much less than "S_a," and similarly, $\dfrac{S_e}{E_eE_gE_{cm}}$ less than "S_c." In other words, the steam consumption of

the main engine per I. H. P. hour, divided by the combined efficiency of engine, generator, and auxiliary motor, will be less than that of the corresponding auxiliary, should it be steam driven, and this is exactly the advantage claimed for power-driven over steam-driven auxiliaries. On the other hand, there is no exhaust steam available for heating the feed-water so that the factor of evaporation " F " is considerably decreased which will be found to more than offset in most cases the gain in steam consumption. If, however, some other source of waste heat be available for feed-water heating purposes, sufficient in volume and temperature to effect the proper raise in the temperature of the boiler feed, it will be readily seen that the steam consumption of the plant will be reduced, while the factor " F " will remain at its maximum with a resulting economy figure in favor of power-driven auxiliaries. This is exactly what occurs when motor or belt-driven air and circulating pumps are installed in connection with fuel economizers.

TABLES AND GRAPHS

As a convenience in solving equation (68) the accompanying figures may be used. The numerator will, of course, be solved exactly as in the preceding equations. Figure 28 shows in graphical form values of $\dfrac{W h S_e}{792\, E_e E_g E_c E_{cm}}$ for assumed values of " W_h," " S_e," and " $E_e E_g$," and and " $E_c E_{cm}$." This graph is used in an exactly similar manner to Fig. 27. Starting with the proper value of " Wh " at the bottom of the sheet, move vertically until reaching the line denoting the required figure for " S_e," thence horizontally until one of the " $E_e E_g$ " lines is encountered, bearing the proper denomination, thence vertically again until reaching " $E_c E_{cm}$ " as desired, and, finally, horizontally to the left-hand side of the graph, where the required value of $\dfrac{W h S_e}{792\, E_e E_g E_c E_{cm}}$ is found.

Figure 29 tabulates results of $\dfrac{S_e}{E_e E_g E_{am}}$ for assumed values of " S_e," " $E_e E_g$ " and " E_{am}." In this figure " S_e " is first located on the upper margin, from which point a vertical line is dropped until reaching one of the lines denoting the correct value of " $E_e E_g$." From this point move horizontally until reaching the proper value for " E_{cm} " when the required denomination of the fraction $\dfrac{S_e}{E_e E_g E_{cm}}$ will be found immediately below on the lower margin.

Having selected from the tables the proper valves for these two expressions, the economy of the plant may be read directly from Fig. 30, which is used in a similar manner to Fig. 27 for steam-driven auxiliaries.

To illustrate the use of these tables, and at the same time to afford

an actual comparison between equations (53) and (68), the latter formula will be solved under somewhat similar conditions to those above selected for the former.   To this end, the following values will be assumed.

FIG. 28. — VALUES OF $\dfrac{\text{``}WhS_e\text{''}}{792\,E_eE_gE_cE_{cm}}$ — (FOR SURFACE-CONDENSING PLANTS — POWER-DRIVEN AUXILIARIES)

Numerator of

equation (53).. 10,000,000

FS_e 19.6

$\dfrac{S_e}{E_e E_g E_{am}}$ .... 25.

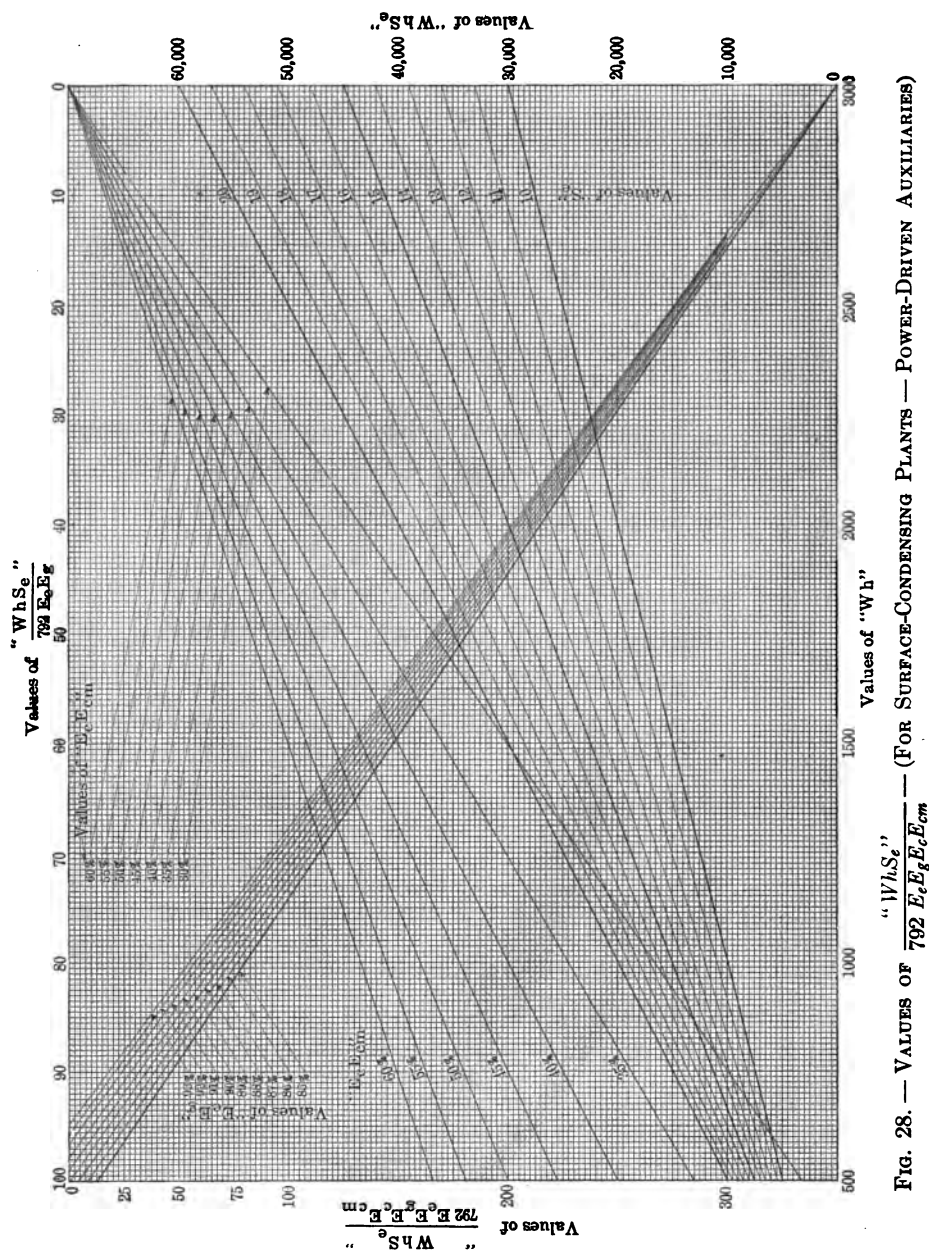$\dfrac{WhS_e}{792\ E_e E_g E_c E_{cm}}$ .... 20.

FIG. 29. — VALUES OF $\dfrac{S_e}{E_e E_g E_{cm}}$ (FOR SURFACE-CONDENSING PLANTS — POWER-DRIVEN AUXILIARIES)

A word is necessary regarding these assumptions and their relation to those previously made in solving equation (53). In the latter case, the feed-water was assumed to be at about 200° Fahr. temperature, and FS_e was taken at 18. With power-driven auxiliaries and a high vacuum, the resulting temperature of the feed-water after being heated by the steam from the feed-pump might be not more than 110° which changes the factor of evaporation, and hence "FS_e." This is a measure of the economic gain realized by heating the feed-water. The assumed values 25 and 20 are comparable with 40 and 50 as taken in equation (53) and show the gain in steam consumption due to the superior economy of the main engine over that of the steam-driven auxiliaries.

But to apply the assumed values to the equation — first locate 20 upon the lower edge of Fig. 30, move vertically until reaching one of the parallel lines representing $\dfrac{S_e}{E_e E_g E_{am}}$ equal to 25, thence horizontally until the line denoting "FS_e" $= 19.6$ is encountered (which must be interpolated for), again vertically to the curved line showing the numerator of the fraction as 10,000,000, and finally horizontally to the left-hand margin where the final result is found to be about 188 Kw. hours per barrel of oil. It will be remembered that the plant having steam-driven auxiliaries figured about 208 Kw. hours — a very considerable gain. The assumptions, however, made in equation (68) are hardly as favorable as those embodied in (53) so that the comparison is more or less forced.

Methods have been outlined above, and tables and graphs presented for solving —

    (1) Non-condensing plants.
    (2) Surface-condensing plants — steam-driven auxiliaries.
    (3) Surface-condensing plants — power-driven auxiliaries.

A similar method may be outlined for any required combination of auxiliaries — such as the wet and dry vacuum system or power-driven feed-pumps, for instance, — but these figures must be left to the ingenuity of the investigator, as the complete representation of every conceivable combination of the different types of pumps would entail an endless amount of work, would require unlimited space, and would practically be useless. An attempt has been made, therefore, to select such arrangements as have become more or less "standard," leaving special cases for individual and outside investigation. Jet-condensing plants will now be considered along lines similar to the above.

JET–CONDENSING PLANTS

In a jet-condensing plant, the air-pump replaces the air and circulating pumps, in a measure, of a surface-condensing plant, inasmuch as it handles

both the condensed steam and the injection water. It is usually necessary, however, to obtain the latter under sufficient pressure from some outside source, and for this purpose either a centrifugal or direct acting pump is commonly employed. This pump we will designate the "injection water-pump." In order to retain a simplicity of notation and, further, in view of the striking similarity between the duty of the injection water pump and that of the circulating pump in surface-condensing plants, the same system of notation has been employed. That is to say, the symbols "W," "h," "S_c," and "E_c" will represent the same values with relation to the injection water-pump in a jet-condensing plant, as they do with relation to the circulating pump in a surface-condensing plant.

Feed, air, and injection water-pumps will all be assumed steam-driven in accordance with usual practice. The difference between crank and flywheel and direct-acting air-pumps, or engine-driven centrifugal and direct-acting injection water-pumps, will appear when the proper numerical values are applied in the formula for their steam consumptions and mechanical efficiencies. The algebraic process is as follows:

From equation (6) we get the steam consumption of the main engine in pounds per hour;

$$\frac{KS_e}{.746\,E_eE_g}$$

The air-pump is next to be considered. The quantity of water pumped per hour by this auxiliary, allowing both for injection water and condensed steam, will be,

$$\frac{KS_e\,(W+1)}{.746\,E_eE_g} \qquad \cdots \quad \cdots \quad \cdots \quad (69)$$

and the head pumped against in feet (assuming 26-inch vacuum and a discharge lift of ten feet)

$$\frac{26\times 14.7\times 2.31}{30} + 10 = 39.43, \text{ or say 40 feet,}$$

from which the steam consumption of the air-pump in pounds per hour, assuming a pump efficiency of 70%, will equal,

$$\frac{40\,KS_e\,(W+1)\,S_a}{33,000\times 60\times .746\,E_eE_g\times .7} = \frac{KS_eS_a\,(W+1)}{99,000\times .746\times .35\,E_eE_g}. \quad (70)$$

Similarly the injection water-pump will consume,

$$\frac{KS_eWhS_c}{60\times 33,000\times .746\,E_eE_gE_c} \qquad \cdots \quad \cdots \quad (71)$$

pounds of steam per hour.

Adding (6), (70) and (71), (the steam consumption of main engine, air-pump and injection water-pump) and reducing, we have,

$$\frac{KS_e\,[33,000 \times 60\,E_c + 57.14\,E_cS_a\,(W+1) + \ WhS_c]}{60 \times 33,000 \times .746\,E_cE_eE_g} \qquad . \quad (72)$$

It now becomes necessary to estimate the steam consumption per hour of the feed-pump, which is equal to

$$p = \frac{\left\{ \dfrac{KS_e\,[33,000 \times 60\,E_c + 57.14\,E_cS_a\,(W+1) + WhS_c]}{.93 \times 60 \times 33,000 \times .746\,E_cE_eE_g} + \dfrac{p}{.93} \right\} 2.31\,PS_f}{.93 \times 60 \times 33,000 \times .8} \qquad (73)$$

from which,

$$p = \frac{.07\,PS_f\,KS_e\,[60 \times 33,000\,E_c + 57.14\,E_cS_a\,(W+1) + Whs_c]}{60 \times 33,000 \times .746\,E_cE_eE_g\,(.93 \times 20 \times 3000 \times .8 - .07\,PS_f)} \qquad 74)$$

Adding (72) and (74), dividing the sum by .93 to allow for other auxiliaries and losses as explained above, and multiplying the quotient by "F," the total steam consumption of the entire plant in pounds per hour from and at 212° Fahr., will equal:

$$\frac{.93 \times 20 \times 3000 \times .8\,KFS_e\,[60 \times 33,000\,E_c + 57.14\,E_cS_a\,(W+1) + WhS_c]}{.93 \times 60 \times 33,000 \times .746\,E_cE_eE_g\,(.93 \times 20 \times 3000 \times .8 - .07\,PS_f)} \qquad (75)$$

Dividing (75) by $\dfrac{336\,HE_b}{966}$ (the equivalent evaporation per barrel of oil) we have the barrels of oil consumed per hour to be

$$\frac{23\,KFS_e\,[60 \times 33,000\,E_c + 57.14\,E_cS_a\,(W+1) + WhS_c]}{330 \times .746\,HE_bE_eE_gE_c\,(.93 \times 20 \times 3000 \times .8 - .07\,PS_f)} \qquad . \quad (76)$$

The kilowatt hours output per barrel of oil burned may now be obtained by dividing the kilowatts developed in one hour by the fuel consumed in that period of time, thus: Kw. hours per barrel of oil equal,

$$K_h = \frac{330 \times .746\,HE_bE_eE_gE_c\,(.93 \times 20 \times 3000 \times .8 - .07\,PS_f)}{23\,FS_e\,[60 \times 33,000\,E_c + 57.14\,E_cS_a\,(W+1) + WhS_c]} \qquad (77)$$

Reducing,

$$K_h = \frac{246.18\,HE_bE_eE_gE_c\,(44,640 - .07\,PS_f)}{23\,FS_e\,[60 \times 33,000\,E_c + 57.14\,E_cS_a\,(W+1) + WhS_c]} \qquad (78)$$

and

$$K_h = \frac{7 \times 41.03\,HE_bE_eE_g\left(638 - \dfrac{PS_f}{1000}\right)}{23FS_e\left[33,000 + S_a\,(W+1) + \dfrac{WhS_c}{60\,E_c}\right]} \qquad . \quad . \quad . \quad (79)$$

Fig. 31.—Values of $\dfrac{\text{``}WhS_c\text{''}}{60\,E_c}$ (For Jet-Condensing Plants)

"S_c"	Values of "Wh"						
	500	700	1000	1500	2000	2500	3000
"E_c" = 40%							
30	625.00	875.00	1250.00	1875.00	2500.00	3125.00	3750.00
40	833.33	1166.67	1666.67	2500.00	3333.33	4166.67	5000.00
50	1041.67	1458.33	2083.33	3125.00	4166.67	5208.33	6250.00
60	1250.00	1750.00	2500.00	3750.00	5000.00	6250.00	7500.00
100	2083.33	2916.67	4166.67	6250.00	8333.33	10416.67	12500.00
150	3125.00	4375.00	6250.00	9375.00	12500.00	15625.00	18750.00
"E_c" = 45%							
30	555.56	777.78	1111.11	1666.67	2222.22	2777.78	3333.33
40	740.74	1037.04	1481.48	2222.22	2962.96	3703.70	4444.44
50	925.93	1296.30	1851.85	2777.78	3703.70	4629.63	5555.56
60	1111.11	1555.56	2222.22	3333.33	4444.44	5555.56	6666.67
100	1851.85	2592.59	3703.70	5555.56	7407.41	9259.26	11111.11
150	2777.78	3888.89	5555.56	8333.33	11111.11	13888.89	16666.67
"E_c" = 50%							
30	500.00	700.00	1000.00	1500.00	2000.00	2500.00	3000.00
40	666.67	933.33	1333.33	2000.00	2666.67	3333.33	4000.00
50	833.33	1166.67	1666.67	2500.00	3333.33	4166.67	5000.00
60	1000.00	1400.00	2000.00	3000.00	4000.00	5000.00	6000.00
100	1666.67	2333.33	3333.33	5000.00	6666.67	8333.33	10000.00
150	2500.00	3500.00	5000.00	7500.00	10000.00	12500.00	15000.00
"E_c" = 55%							
30	454.55	636.36	909.09	1363.64	1818.18	2272.73	2727.27
40	606.06	848.48	1212.12	1818.18	2424.24	3030.30	3636.36
50	757.58	1060.61	1515.15	2272.73	3030.30	3787.88	4545.45
60	909.09	1272.73	1818.18	2727.27	3636.36	4545.45	5454.55
100	1515.15	2121.21	3030.30	4545.45	6060.60	7575.76	9090.91
150	2272.73	3181.82	4545.45	6818.18	9090.91	11363.64	13636.36
"E_c" = 60%							
30	416.67	583.33	833.33	1250.00	1666.67	2083.33	2500.00
40	555.56	777.78	1111.11	1666.67	2222.22	2777.78	3333.33
50	694.44	972.22	1388.89	2083.33	2777.78	3422.22	4166.67
60	833.33	1166.67	1666.67	2500.00	3333.33	4166.67	5000.00
100	1388.89	1944.44	2777.78	4166.67	5555.56	6944.44	8333.33
150	2083.33	2916.67	4166.67	6250.00	8333.33	10416.67	12500.00
"E_c" = 80%							
30	312.50	437.50	625.00	937.50	1250.00	1562.50	1875.00
40	416.67	583.33	833.33	1250.00	1666.67	2083.33	2500.00
50	520.83	729.17	1041.67	1562.50	2083.33	2604.17	3125.00
60	625.00	875.00	1250.00	1875.00	2500.00	3125.00	3750.00
100	1041.67	1458.33	2083.33	3125.00	4166.67	5208.33	6250.00
150	1562.50	2187.50	3125.00	4687.50	6250.00	7812.50	9375.00

whence,

$$K_k = \frac{HE_bE_eE_g\left(638 - \dfrac{PS_f}{1000}\right)}{.08\,FS_e\left[33{,}000 + S_a\,(W + 1) + \dfrac{WhS_c}{60\,E_c}\right]} \qquad . \quad . \quad (80)$$

Equation (80) is final and correct for all jet-condensing plants using steam-driven auxiliaries. In cases where it is desired to figure the economy of such a plant, omitting the injection water-pump (if, for instance, sufficient pressure for the injection is available from other sources), it is only necessary to cancel the last term in the denominator of the above fraction.

TABLES AND GRAPHS FOR JET-CONDENSING PLANTS

As a matter of interest, it will be noted that in all equations for condensing plants the numerator takes exactly the same form. In applying equation (80) the numerator may be obtained from Figs. 21 and 24 and "FS_e" from Fig. 22, all as explained above. Figure 31 tabulates the expression $\dfrac{WhS_c}{60\,E_c}$ and Fig. 32 the expression "$S_a\,(W + 1)$." The final result is read from the chart, Fig. 33, which is used precisely as Figs. 27 and 30 for surface-condensing plants.

It has been deemed unnecessary to investigate jet-condensing plants having motor or belt-driven pumps as this combination is very unusual for Pacific Coast conditions. Suffice it to say that the same method could be applied to this or any other arrangement with equal facility. Pumping plants will be briefly discussed in the pages to follow.

PUMPING PLANTS

In any steam plant, designed for pumping water, steam is consumed by:
   (1) The engine operating the main pump or pumps.
   (2) The auxiliaries, including oil burners, radiation, leakage, etc.

The quantity of steam required by the main pumping engine depends upon the economy of the particular type of machine under consideration and the amount of power required by the pump. The latter value is determined by the amount of water to be pumped, the total pressure at which the discharge occurs, and the efficiency of the pump.

The steam required by the auxiliaries is, in general, similar to that for a power or lighting plant, except that in surface-condensing pumping plants, a circulating pump is but seldom required, owing to the large quantity of water under pressure available from the main pumps. In case the water pumped, however, is not suitable for condensing purposes,

or in other special cases, a circulating pump is required. The following figures will cover non-condensing and surface-condensing plants, in which circulating pumps are not necessary.

The same system of notation will be employed as in the demonstrations covering electric plants above, and in addition the following:
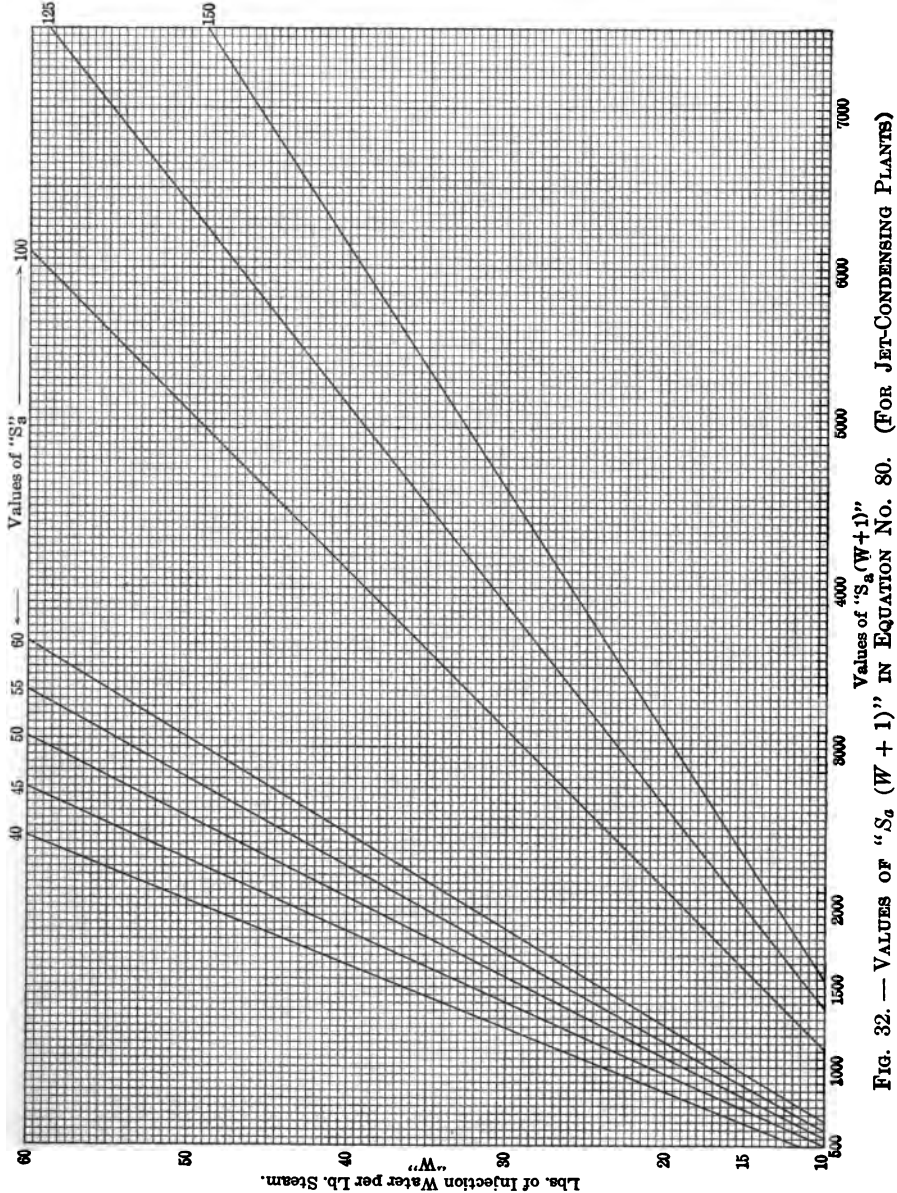

Fig. 32. — Values of "$S_a (W + 1)$" in Equation No. 80. (For Jet-Condensing Plants)

G = capacity of main pump in U. S. gallons of water per minute.

T = total head pumped against including friction, expressed in feet.

D_p = duty of main pumping engine expressed in millions of foot pounds per 1000 lbs. steam at the throttle.

D = duty of complete plant expressed in millions of foot pounds per barrel of oil consumed.

E_p = efficiency of main pump.

There are two general classes of steam-driven pumping plants; — (1) those in which a steam engine is belted or direct connected to some form of pump (usually either a centrifugal or triplex arrangement), in which the pump simply takes the place, in a measure, of the generator in an electric plant; and (2) those in which the engine and pump are combined in one machine forming a crank and fly-wheel high duty pumping engine or a direct acting pump. Each of these is solved by a separate method and these two methods are sufficient for any type of steam-driven pumping plant. It will be the object of the figures to follow, to arrive at an expression representing "D" in terms of the known variables entering into the economical performance of both types of plants.

For convenience, the first type will be called "centrifugal or triplex pumping plants," and the second, "high duty or direct acting pumping plants."

CENTRIFUGAL OR TRIPLEX PUMPING PLANTS

The efficiency of a centrifugal pump depends largely upon its size, the head against which it operates, the speed and diameter of the runner, and the curvature of the vanes. It is practically impossible, therefore, to tabulate these efficiencies for all conditions, and any attempt to do so would necessarily result in more confusion than real value. Some large pumps have shown very high efficiencies at low heads when designed with small runners for high speeds, but in general it is easier to obtain the better efficiencies with heads of twenty feet or over. Open runner pumps are always extremely inefficient. With the very small sizes, 40% seems to be about the average; while with large, well-designed enclosed runner pumps, over 75% has been obtained. The average reader has fairly well in mind the results to be obtained under different conditions, but it is always well to carefully study these conditions, or, better still, to obtain guarantees from the manufacturers before selecting the final figure to be used.

.With triplex pumps, on the other hand, the efficiencies are known quantities. As a general rule, the greater the head pumped against, the higher the efficiency within limits, for the reason that the work lost in overcoming the friction in the pump itself (being practically constant), represents a smaller portion of the total work required to pump the water the greater the head becomes. Figure 34 represents the efficiency of well-designed triplex pumps at different heads.

The same formula and method of discussion may, however, be used regardless of whether the pump be centrifugal or triplex.

The indicated horse-power developed by the main engine will be —

$$\frac{\frac{25}{3} G \times T}{33,000 \, E_e E_p} = \frac{25 \, GT}{99,000 \, E_e E_p} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (81)$$
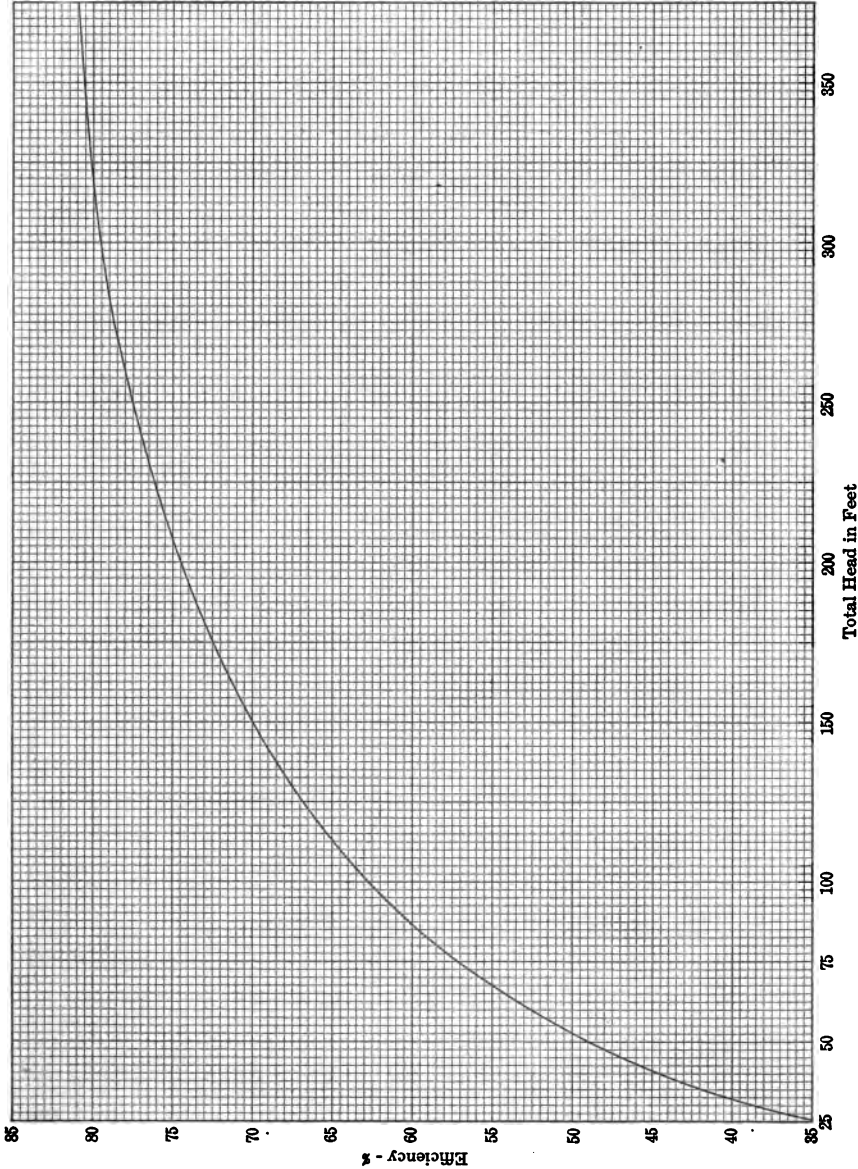
FIG. 34. — EFFICIENCY OF TRIPLEX PUMPS

from which the steam consumed per hour by the main engine will be,

$$\frac{25\,GTS_e}{99{,}000\,E_eE_p} \qquad \cdots \quad \cdots \quad \cdots \tag{82}$$

The steam consumption of the air-pump per hour will equal

$$\frac{.00024 \times 25\,GTS_eS_a}{.6 \times 99{,}000\,E_eE_p} = \frac{GTS_eS_a}{9{,}900{,}000\,E_eE_p} \qquad \cdots \quad \cdots \tag{83}$$

Adding (82) and (83) we have the steam required per hour by the engine and air-pump, viz.:

$$\frac{GTS_e\,(25 + .01\,S_a)}{99{,}000\,E_eE_p} \qquad \cdots \quad \cdots \quad \cdots \tag{84}$$

Solving for the steam used by the feed-pump in general accordance with the method outlined under electric plants, we have,

$$p = \frac{\left[\dfrac{GTS_e\,(25 + .01\,S_a)}{.93 \times 99{,}000\,E_eE_p} + \dfrac{p}{.93}\right]2.31\,PS_f}{60 \times 33{,}000 \times .8} \qquad \cdots \quad \cdots \tag{85}$$

from which

$$p = \frac{2.31\,PS_fGTS_e\,(25 + .01\,S_a)}{99{,}000\,E_eE_p\,(60 \times .8 \times .93 \times 33{,}000 - 2.31\,PS_f)} \qquad \cdots \quad \cdots \tag{86}$$

Adding (84) and (86), dividing by .93 for other auxiliaries and losses, and multiplying by "F," we have the total steam required by the plant per hour from and at 212° Fahr., equals:

$$\frac{16\,FGTS_e\,(25 + .01\,S_a)}{E_eE_p\,(60 \times .8 \times .93 \times 33{,}000 - 2.31\,PS_f)} \qquad \cdots \quad \cdots \tag{87}$$

Equation (87) divided by the equivalent evaporation per barrel of oil "$\dfrac{8\,HE_b}{23}$" will give the fuel consumption of the entire plant in barrels per hour, which is —

$$\frac{46\,FGTS_e\,(25 + .01\,S_a)}{HE_bE_eE_p\,(60 \times .8 \times .93 \times 33{,}000 - 2.31\,PS_f)} \qquad \cdots \quad \cdots \tag{88}$$

In order, now, to express the duty of the entire plant in terms of millions of foot pounds of work performed per barrel of oil burned ("D"), it will be necessary only to obtain the number of millions of foot pounds performed per hour, and to divide this quantity by equation (88). The number of millions of foot pounds performed per hour equals

$$\frac{60 \times 25\,GT}{3 \times 1{,}000{,}000} \qquad \cdots \quad \cdots \quad \cdots \tag{89}$$

"D" - Duty in Terms of Millions of Foot Pounds per Bbl. Oil

Values for Numerator of Equation No. 90 - "2.61 H.E_b E_e E_p $\left(638 - \dfrac{PS_f}{1000}\right)$"
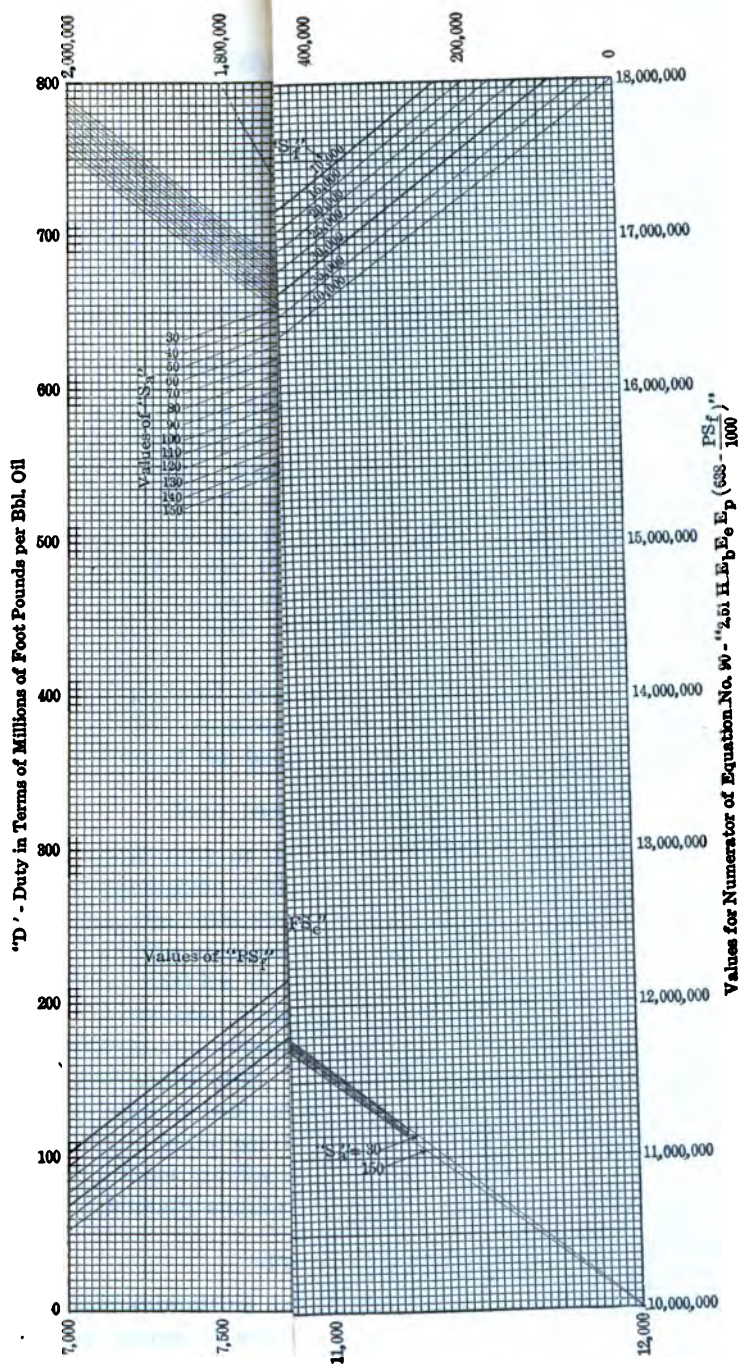
PUMPING PLANTS)

which, divided by (88) and reduced, equals —

$$D = \frac{2.51\, HE_bE_eE_p \left(638 - \dfrac{PS_f}{1000}\right)}{FS_e\,(2500 + S_a)} \quad . \quad . \quad . \quad . \quad . \quad (90)$$

Any engine-driven centrifugal or triplex pumping plant may be solved by substituting the proper numerical values in equation (90). It will be noticed that this equation is of the same form as those for the electric plants developed above.

TABLES AND GRAPHS FOR CENTRIFUGAL OR TRIPLEX PUMPING PLANTS

In applying this formula, the proper numerical value of "$HE_bE_eE_p$" may be obtained from Fig. 38 in exactly the same manner as in the case of electric plants, having obtained which, together with the required value for "FS_e," the result may be read directly from Fig. 35. In using this chart, locate the proper value of "$HE_bE_eE_p$" on the lower margin, move vertically until reaching the required value of "PS_f," thence horizontally to one of the lines denoting the proposed denomination of "FS_e," again vertically until one of the closely drawn lines denoting values of "S_a" is reached, and finally horizontally to the left-hand vertical scale where the required result may be read. In case the pump be belted instead of direct connected to the engine, care should be taken to include the efficiency of the belt drive with that of the pump or engine.

HIGH DUTY OR DIRECT-ACTING PUMPING PLANTS

In a high duty or direct acting pumping plant as defined above, the steam consumption of the engine alone is not usually taken as a basis for estimating the total steam required by the plant, but rather the duty of the pumping engine expressed in millions of foot pounds of work performed per 1000 lbs. of steam delivered to the throttle. This figure may always be obtained from the pump manufacturer and includes the steam consumption and efficiency of the engine end, and the efficiency of the pump end of the unit. The calculation is therefore much simplified, the method being as hereinafter set forth, to estimate the steam consumption of the entire plant per 1000 lbs. at the engine instead of per hour as in preceding cases, having obtained which, the remainder of the process remains practically similar to that above outlined.

PUMPING ENGINE ECONOMY

Before submitting actual equations for high duty or direct acting pumping plants, it will be well to say a few words regarding the duty and efficiency of these machines. Starting with single cylinder duplex steam pumps, the duty ranges roughly between 10,000,000 foot pounds

for the small unjacketed non-condensing pumps to 50,000,000 foot pounds for larger compound duplex condensing arrangements. Figure 36 shows graphically the duties to be expected from duplex pumps, both simple and compound under 100 lbs. steam pressure at the throttle, and discharging against a head of 150 lbs. These figures should be safe for test con-
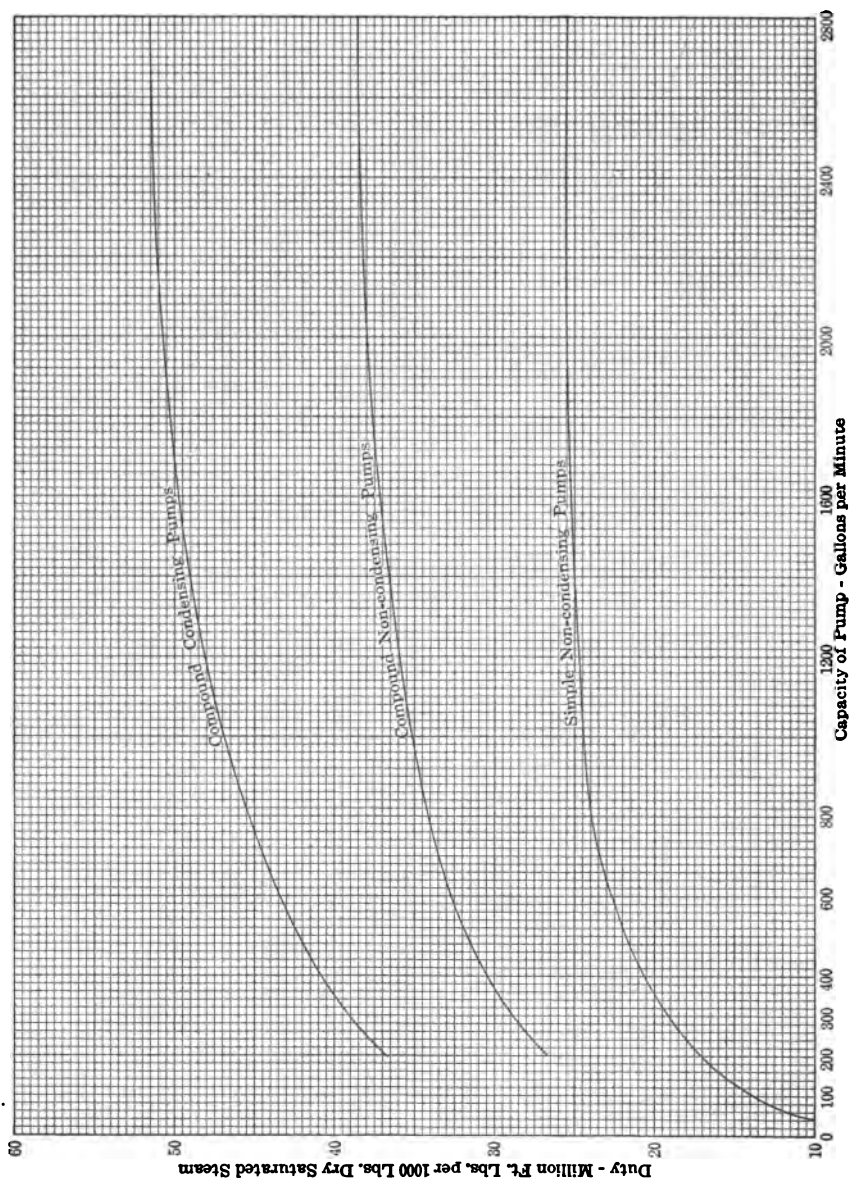
FIG. 36. — DUTY OF DUPLEX DIRECT ACTING PUMPS. BASED ON 100 LBS. I. S. P. 80 TO 90 FEET PISTON SPEED AND 150 LBS. WATER PRESSURE

ditions, care being taken in selecting the capacity, that the pump is of such size as to easily pump the selected quantity of water at a piston speed not exceeding 80 or 90 ft. per minute. Duties for other conditions of steam pressure, water pressure, etc., may be approximated from the above, care being taken, however, to fully analyze the conditions of operation, and their effect upon the duty before making final selection.

The efficiency of these duplex pumps should reach under good conditions 85% easily for the larger sizes and 80% for the smaller.

Owing to the numerous conditions and the almost unlimited combinations of conditions under which high duty crank and fly-wheel pumping engines operate, it is a matter of great difficulty to present in a compact form any reliable data upon the economy of these machines. Much also depends upon the degree of excellence of the design and workmanship and upon the make of each engine. All that can be done is to present a chart showing results that have been obtained under an assumed "standard" set of conditions, which must be taken as an adequate guide for present purposes.

Accordingly, Fig. 37 sets forth duties for cross-compound crank and fly-wheel Corliss gear pumping engines operating condensing with 26-inch effective vacuum and 150 lbs. steam pressure at the throttle. Results are plotted in terms of millions of foot pounds per 1000 lbs. of dry saturated steam at the above specified pressure. As will be noted figures are based upon a machine having a capacity of 10,000,000 U. S. gallons per 24 hours. In the upper right-hand corner will be found a small curve giving relative percentages for engines of varying sizes. The product of the duty of the 10,000,000 gallon pump and the proper "relative percentage" will give the required duty of a pumping engine of corresponding size. All figures are based upon a rotative speed of not exceeding 40 or 45 revolutions per minute, and a moderate piston speed of 275 ft. per minute or less. It is very seldom, if ever, that engines of this type are operated non-condensing, so figures on this basis aside from being unobtainable would be practically worthless.

Should superheated steam (100° Fahr.) be used, these figures may be increased about 10% and if Meyer gear engine ends, or any similarly geared engine be under consideration, figures should be reduced by from 15% to 20%.

With the above results in mind, formulæ will be developed first for non-condensing plants (which will apply mostly to small direct-acting pumps) and secondly to condensing plants.

NON-CONDENSING PLANTS

In a non-condensing plant of the type under consideration, steam is used or consumed: (1) by the main pumping engine; (2) by the feed-

pumps; (3) by the oil-pumps; (4) by the oil burners; (5) by radiation and leakage. As in all previous examples, the last three items represent a total of 7% of the plant consumption.

Fig. 37. — Duty of Crank and Fly-wheel Pumping Engines. Based on Corliss Compound Condensing Engines, 150 Lbs. I. S. P.

FIG. 38: VALUES OF "$HE_bE_eE_p$." (FOR PUMPING PLANTS)

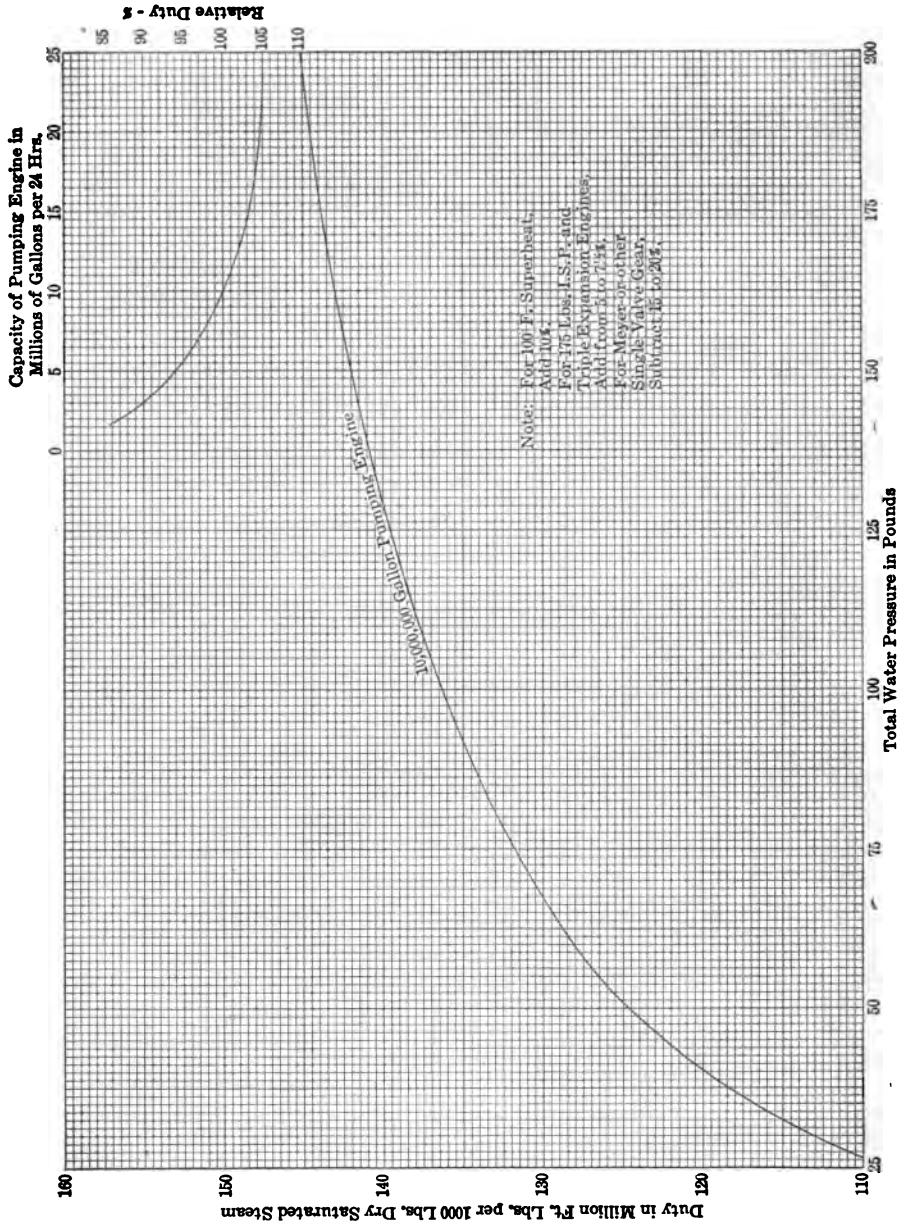| "E_b" %	VALUES OF "E_eE_p" — %															
	50%	52	54	56	58	60	62	64	66	68	70	72	74	76	78	80%
	"H" = 18000															
70	6300	6552	6804	7056	7308	7560	7812	8064	8316	8568	8820	9072	9324	9576	9828	10080
71	6390	6646	6901	7157	7412	7668	7924	8179	8435	8690	8946	9202	9457	9713	9968	10224
72	6480	6739	6998	7258	7517	7776	8035	8294	8554	8813	9072	9331	9590	9850	10109	10368
73	6570	6833	7096	7358	7621	7884	8145	8410	8672	8935	9198	9461	9724	9986	10249	10512
74	6660	6926	7193	7459	7726	7992	8258	8525	8791	9058	9324	9590	9857	10123	10390	10656
75	6750	7020	7290	7560	7830	8100	8370	8640	8910	9180	9450	9720	9990	10260	10530	10800
76	6840	7114	7387	7661	7934	8208	8482	8755	9029	9302	9576	9850	10123	10397	10670	10944
77	6930	7207	7484	7762	8039	8316	8593	8870	9148	9425	9702	9979	10256	10534	10811	11088
78	7020	7301	7582	7862	8143	8424	8705	8986	9266	9547	9828	10109	10390	10670	10951	11232
79	7110	7394	7679	7963	8248	8532	8816	9101	9385	9670	9954	10238	10523	10807	11092	11376
80	7200	7488	7776	8064	8352	8640	8928	9216	9504	9792	10080	10368	10656	10944	11232	11520
81	7290	7582	7873	8165	8456	8748	9040	9331	9623	9914	10206	10498	10789	11081	11372	11664
82	7380	7675	7970	8266	8561	8856	9151	9446	9742	10037	10332	10627	10922	11218	11513	11808
83	7470	7769	8068	8366	8665	8964	9263	9562	9860	10159	10458	10757	11056	11354	11653	11952
	"H" = 18500															
70	6475	6734	6993	7252	7511	7770	8029	8288	8547	8806	9065	9324	9583	9842	10101	10360
71	6567	6830	7093	7356	7618	7881	8144	8406	8669	8932	9195	9457	9720	9983	10245	10508
72	6660	6926	7193	7459	7726	7992	8259	8525	8791	9058	9324	9590	9857	10123	10390	10656
73	6752	7023	7293	7563	7833	8103	8373	8643	8913	9183	9453	9724	9994	10264	10534	10804
74	6845	7119	7393	7666	7940	8214	8488	8762	9035	9309	9583	9857	10131	10404	10678	10952
75	6937	7215	7493	7770	8047	8325	8603	8880	9157	9435	9712	9990	10267	10545	10822	11100
76	7030	7311	7592	7874	8155	8436	8718	8998	9280	9561	9842	10123	10404	10686	10967	11248
77	7122	7407	7692	7977	8262	8547	8832	9117	9402	9687	9971	10256	10541	10826	11111	11396
78	7215	7504	7792	8081	8369	8658	8947	9235	9524	9812	10101	10390	10678	10967	11255	11544
79.	7307	7600	7892	8184	8477	8769	9062	9354	9646	9938	10230	10523	10815	11107	11400	11692
80	7400	7696	7992	8288	8584	8880	9176	9472	9768	10064	10360	10656	10952	11248	11544	11840
81	7492	7792	8092	8392	8691	8991	9291	9590	9890	10190	10489	10789	11089	11389	11688	11988
82	7585	7889	8192	8495	8799	9102	9406	9709	10012	10316	10619	10922	11226	11529	11833	12136
83	7678	7985	8292	8599	8906	9213	9520	9827	10134	10441	10749	11056	11363	11670	11977	12284
	"H" = 18850															
70	6598	6861	7125	7389	7653	7917	8181	8445	8709	8973	9237	9500	9763	10028	10292	10556
71	6692	6959	7227	7495	7762	8030	8298	8565	8833	9101	9368	9636	9903	10171	10439	10707
72	6786	7057	7329	7600	7872	8143	8415	8686	8957	9229	9500	9772	10042	10315	10586	10858
73	6880	7155	7431	7706	7981	8256	8532	8807	9082	9357	9632	9908	10182	10458	10733	11008
74	6975	7253	7532	7811	8090	8369	8648	8927	9206	9485	9764	10043	10321	10601	10880	11159
75	7069	7352	7634	7917	8200	8483	8765	9048	9331	9614	9896	10179	10461	10745	11027	11310
76	7163	7450	7736	8023	8309	8596	8882	9169	9455	9742	10028	10315	10600	10888	11174	11461
77	7257	7548	7838	8128	8418	8709	8999	9289	9579	9870	10160	10450	10740	11031	11321	11612
78	7352	7646	7940	8234	8528	8821	9116	9410	9704	9998	10292	10586	10879	11174	11468	11762
79	7446	7744	8041	8339	8637	8935	9233	9531	9828	10126	10424	10722	11019	11318	11615	11913
80	7540	7842	8143	8445	8746	9048	9350	9651	9953	10254	10556	10858	11158	11461	11762	12064
81	7634	7940	8245	8550	8855	9161	9466	9772	10077	10383	10688	10993	11298	11604	11909	12215
82	7729	8038	8347	8656	8965	9274	9583	9892	10202	10511	10820	11129	11437	11747	12056	12366
83	7823	8136	8449	8761	9074	9387	9700	10013	10326	10639	10952	11265	11578	11891	12203	12516

The engine, in order to perform "D_p" foot pounds, requires 1000 lbs. of steam.   The feed-pump requires;

$$p = \frac{\left[\dfrac{1000}{.93} + \dfrac{p}{.93}\right] 2.31 \, PS_f}{60 \times 33,000 \times .8} \qquad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (91)$$

from which

$$p = \frac{2310 \, PS_f}{(1,473,120 - 2.31 \, PS_f)} \qquad \cdot \quad \cdot \quad \cdot \quad : \quad \cdot \quad (92)$$

Therefore the total equivalent steam used by the entire plant from and at 212° Fahr. per 1000 lbs. at the main engine equals

$$F\left[\frac{1000}{.93} + \frac{2310 \, PS_f}{.93 \, (1,473,120 - 2.31 \, PS_f)}\right] \qquad \cdot \quad \cdot \quad \cdot \quad (93)$$

which reduced, gives —

$$\frac{1,473,120,000 \, F}{.93 \, (1,473,120 - 2.31 \, PS_f)} \qquad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (94)$$

Dividing by $\dfrac{8 \, HE_b}{23}$ gives the barrels of oil burned under the boilers per 1000 lbs. steam supplied the main engine, which is

$$\frac{23 \times 1,473,120,000 \, F}{.93 \times 8 \, HE_b \, (1,473,120 - 2.31 \, PS_f)} \qquad \cdot \quad \cdot \quad \cdot \quad (95)$$

But equation (95) also represents the barrels of oil burned in order to obtain the duty "D_p," from which it follows that the ratio of "D_p" and equation (95) will represent the required value of "D," thus:

$$D = \frac{.93 \times 8 \, HE_b D_p \, (1,473,120 - 2.31 \, PS_f)}{23 \times 1,473,120,000 \, F} \qquad \cdot \quad \cdot \quad \cdot \quad (96)$$

Reducing and simplifying,

$$D = \frac{D_p HE_b \left(638 - \dfrac{PS_f}{1000}\right)}{1,971,430 \, F} \qquad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (97)$$

TABLES AND GRAPHS FOR NON-CONDENSING HIGH DUTY OR DIRECT-ACTING PUMPING PLANTS

In applying equation (97), Figs. 39 and 40 can be used to advantage. Figure 39 shows in graphical form values of "$D_p HE_b$" for various values of "D_p" and "E_b" and for "H" = 18,000, 18,500 and 18,850.   Figure

FIG. 39. — VALUES OF "$D_p H E_b$" (FOR PUMPING PLANTS)

VALUES OF "D_p"

"H" = 18000

"E_b" %	20	25	30	35	40	45	50	80	85	90	95	100	105	110	115	120	125	130	135	140	145	150
70	252000	315000	378000	441000	504000	567000	630000	1008000	1071000	1134000	1197000	1260000	1323000	1386000	1449000	1512000	1575000	1638000	1701000	1764000	1827000	1890000
71	255600	319500	383400	447300	511200	575100	639000	1022400	1086300	1150200	1214100	1278000	1341900	1405800	1469700	1533600	1597500	1661400	1725300	1789200	1853100	1917000
72	259200	324000	388800	453600	518400	583200	648000	1036800	1101600	1166400	1231200	1296000	1360800	1425600	1490400	1555200	1620000	1684800	1749600	1814400	1879200	1944000
73	262800	328500	394200	459900	525600	591300	657000	1051200	1116900	1182600	1248300	1314000	1379700	1445400	1511100	1576800	1642500	1708200	1773900	1839600	1905300	1971000
74	266400	333000	399600	466200	532800	599400	666000	1065600	1132200	1198800	1265400	1332000	1398600	1465200	1531800	1598400	1665000	1731600	1798200	1864800	1931400	1998000
75	270000	337500	405000	472500	540000	607500	675000	1080000	1147500	1215000	1282500	1350000	1417500	1485000	1552500	1620000	1687500	1755000	1822500	1890000	1957500	2025000
76	273600	342000	410400	478800	547200	615600	684000	1094400	1162800	1231200	1299600	1368000	1436400	1504800	1573200	1641600	1710000	1778400	1846800	1915200	1983600	2052000
77	277200	346500	415800	485100	554400	623700	693000	1108800	1178100	1247400	1316700	1386000	1455300	1524600	1593900	1663200	1732500	1801800	1871100	1940400	2009700	2079000
78	280800	351000	421200	491400	561600	631800	702000	1123200	1193400	1263600	1333800	1404000	1474200	1544400	1614600	1684800	1755000	1825200	1895400	1965600	2035800	2106000
79	284400	355500	426600	497700	568800	639900	711000	1137600	1208700	1279800	1350900	1422000	1493100	1564200	1635300	1706400	1777500	1848600	1919700	1990800	2061900	2133000
80	288000	360000	432000	504000	576000	648000	720000	1152000	1224000	1296000	1368000	1440000	1512000	1584000	1656000	1728000	1800000	1872000	1944000	2016000	2088000	2160000
81	291600	364500	437400	510300	583200	656100	729000	1166400	1239300	1312200	1385100	1458000	1530900	1603800	1676700	1749600	1822500	1895400	1968300	2041200	2114100	2187000
82	295200	369000	442800	516600	590400	664200	738000	1180800	1254600	1328400	1402200	1476000	1549800	1623600	1697400	1771200	1845000	1918800	1992600	2066400	2140200	2214000

"H" = 18500

"E_b" %	20	25	30	35	40	45	50	80	85	90	95	100	105	110	115	120	125	130	135	140	145	150
70	259000	323750	388500	453250	518000	582750	647500	1036000	1100750	1165500	1230250	1295000	1359750	1424500	1489250	1554000	1618750	1683500	1748250	1813000	1877750	1942500
71	262700	328375	394050	459725	525400	591075	656750	1050800	1116475	1182150	1247825	1313500	1379175	1444850	1510525	1576200	1641875	1707550	1773225	1838900	1904575	1970250
72	266400	333000	399600	466200	532800	599400	666000	1065600	1132200	1198800	1265400	1332000	1398600	1465200	1531800	1598400	1665000	1731600	1798200	1864800	1931400	1998000
73	270100	337625	405150	472675	540200	607725	675250	1080400	1147925	1215450	1282975	1350500	1418025	1485550	1553075	1620600	1688125	1755650	1823175	1890700	1958225	2025750
74	273800	342250	410700	479150	547600	616050	684500	1095200	1163650	1232100	1300550	1369000	1437450	1505900	1574350	1642800	1711250	1779700	1848150	1916600	1985050	2053500
75	277500	346875	416250	485625	555000	624375	693750	1110000	1179375	1248750	1318125	1387500	1456875	1526250	1595625	1665000	1734375	1803750	1873125	1942500	2011875	2081250
76	281200	351500	421800	492100	562400	632700	703000	1124800	1195100	1265400	1335700	1406000	1476300	1546600	1616900	1687200	1757500	1827800	1898100	1968400	2038700	2109000
77	284900	356125	427350	498575	569800	641025	712250	1139600	1210825	1282050	1353275	1424500	1495725	1566950	1638175	1709400	1780625	1851850	1923075	1994300	2065525	2136750
78	288600	360750	432900	505050	577200	649350	721500	1154400	1226550	1298700	1370850	1443000	1515150	1587300	1659450	1731600	1803750	1875900	1948050	2020200	2092350	2164500
79	292300	365375	438450	511525	584600	657675	730750	1169200	1242275	1315350	1388425	1461500	1534575	1607650	1680725	1753800	1826875	1899950	1973025	2046100	2119175	2192250
80	296000	370000	444000	518000	592000	666000	740000	1184000	1258000	1332000	1406000	1480000	1554000	1628000	1702000	1776000	1850000	1924000	1998000	2072000	2146000	2220000
81	299700	374625	449550	524475	599400	674325	749250	1198800	1273725	1348650	1423575	1498500	1573425	1648350	1723275	1798200	1873125	1948050	2022975	2097900	2172825	2247750
82	303400	379250	455100	530950	606800	682650	758500	1213600	1289450	1365300	1441150	1517000	1592850	1668700	1744550	1820400	1896250	1972100	2047950	2123800	2199650	2275500

"H" = 18850

"E_b" %	20	25	30	35	40	45	50	80	85	90	95	100	105	110	115	120	125	130	135	140	145	150
70	263900	329875	395850	461825	527800	593775	659750	1055600	1121575	1187550	1253525	1319500	1385475	1451450	1517425	1583400	1649375	1715350	1781325	1847300	1913275	1979250
71	267670	334588	401505	468423	535340	602258	669175	1070680	1137598	1204515	1271433	1338350	1405268	1472185	1539103	1606020	1672938	1739855	1806773	1873690	1940608	2007525
72	271440	339300	407160	475020	542880	610740	678600	1085760	1153620	1221480	1289340	1357200	1425060	1492920	1560780	1628640	1696500	1764360	1832220	1900080	1967940	2035800
73	275210	344013	412815	481618	550420	619223	688025	1100840	1169643	1238445	1307248	1376050	1444853	1513655	1582458	1651260	1720063	1788865	1857668	1926470	1995273	2064075
74	278980	348725	418470	488215	557960	627705	697450	1115920	1185665	1255410	1325155	1394900	1464645	1534390	1604135	1673880	1743625	1813370	1883115	1952860	2022605	2092350
75	282750	353438	424125	494813	565500	636188	706875	1131000	1201688	1272375	1343063	1413750	1484438	1555125	1625813	1696500	1767188	1837875	1908563	1979250	2049938	2120625
76	286520	358150	429780	501410	573040	644670	716300	1146080	1217660	1289240	1360820	1432400	1503980	1575560	1647140	1718720	1790300	1861880	1933460	2005040	2076620	2148200
77	290290	362863	435435	508008	580580	653153	725725	1161160	1233733	1306305	1378878	1451450	1524023	1596595	1669168	1741740	1814313	1886885	1959458	2032030	2104603	2177175
78	294060	367575	441090	514605	588120	661635	735150	1176240	1249755	1323270	1396785	1470300	1543815	1617330	1690845	1764360	1837875	1911390	1984905	2058420	2131935	2205450
79	297830	372288	446745	521203	595660	670118	744575	1191320	1265778	1340235	1414693	1489150	1563608	1638065	1712523	1786980	1861438	1935895	2010353	2084810	2159268	2233725
80	301600	377000	452400	527800	603200	678600	754000	1206400	1281800	1357200	1432600	1508000	1583400	1658800	1734200	1809600	1885000	1960400	2035800	2111200	2186600	2262000
81	305370	381713	458055	534398	610740	687083	764025	1221480	1297823	1374165	1450508	1526850	1603193	1679535	1755878	1832220	1908563	1984905	2061248	2137590	2213933	2290275
82	309140	386425	463710	540995	618280	695565	772850	1236560	1313845	1391130	1468415	1545700	1622985	1700270	1777555	1854840	1932125	2009410	2086695	2163980	2241265	2318550

40 may be used in obtaining the final numerical value of equation (97) as follows:

(1) Locate upon the upper horizontal scale the proper value of "D_pHE_b." (2) Move downward vertically until encountering one of the oblique lines representing the proper value of "PS_f." (3) From this point move horizontally until reaching the line denoting "F" at its correct value, and (4) read the result (expressed in millions of foot pounds of work performed per barrel of oil burned), immediately below in the lower horizontal scale.

As indicated above, this formula solves non-condensing plants of the high duty or direct-acting type. It is rather an usual case. In a pumping plant, due to the large quantity of circulating water available, condensers are almost invariably installed. In a few cases where simple direct acting pumps are used, condensers are not included, and this formula is designed to meet such conditions. Also, for plants pumping fluids other than water (such as oil, for instance) which could not be used for cooling purposes in a condenser, equation (97) will be found serviceable.

Condensing Plants

In a surface-condensing pumping plant of the type above described, it will be necessary only to take into consideration the steam consumption of the air-pump in addition to the steam used in the non-condensing plant — otherwise these two investigations are identical. It is assumed that circulating pumps will not be used for the same reason as in the case of centrifugal and triplex plants. Figures for surface condensers only will be set forth as these are by far of more frequent occurrence in high-grade installations. The total steam consumption per 1000 lbs. at the main engine may be estimated as follows:

Steam required by the engine . . . . 1000 lbs.
Steam required by the air pump,
$1000 \times .00024 S_a = .24 S_a$ . . . . (98)
Steam required by the feed-pump equals

$$p = \frac{\left[\dfrac{1000 + .24 S_a}{.93} + \dfrac{p}{.93} \right] 2.31 PS_f}{60 \times 33,000 \times .8} \quad \text{. . .} \quad (99)$$

which reduced, gives

$$p = \frac{2.31 PS_f (1000 + .24 S_a)}{1,473,120 - 2.31 PS_f} \quad \text{. . . . .} \quad (100)$$

Adding 1000 to equations (98) and (100), multiplying by "F," and dividing by .93, gives the total equivalent evaporation in the boilers from

and at 212° Fahr. per 1000 lbs. of steam delivered to the main engines, which is,

$$\frac{(1000 + .24 S_a)\ (1,473,120 - 2.31\ PS_f) + (1000 + .24\ S_a)\ 2.31\ PS_f\ F}{(1,473,120 - 2.31\ PS_f)\ .93} \quad (101)$$

which, when simplified, equals —

$$\frac{1,473,120\ F\ (1000 + .24\ S_a)}{.93\ (1,473,120 - 2.31\ PS_f)} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (102)$$

Dividing (102) by $\dfrac{8\ HE_b}{23}$ will give the barrels of oil required for every 1000 lbs. of steam delivered to the main pumping engine, thus —

$$\frac{23 \times 1,473,120\ F\ (1000 + .24\ S_a)}{.93 \times 8\ HE_b\ (1,473,120 - 2.31\ PS_f)} \quad \cdot \quad \cdot \quad \cdot \quad (103)$$

from which the duty of the complete plant expressed in foot pounds per barrel of oil equals

$$D = \frac{.93 \times 8\ D_p HE_b\ (1,473,120 - 2.31\ PS_f)}{23 \times 1,473,120\ F\ (1000 + .24\ S_a)} \quad \cdot \quad \cdot \quad (104)$$

Reducing,

$$D = \frac{D_p HE_b \left(638 - \dfrac{PS_f}{1000}\right)}{1971.43\ F\ (1000 + .24 S_a)} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (105)$$

Formula (105) is correct for high duty or direct-acting pumping plants operating under condensing conditions, circulating water being supplied from the suction or discharge of the main pumping units.

TABLES AND GRAPHS FOR CONDENSING HIGH DUTY OR DIRECT-ACTING PUMPING PLANTS

In solving equation (105) by the aid of the accompanying graphs, first select "$D_p HE_b$" from Fig. 39, knowing which, Fig. 41 may be used in reading the final result. This chart is similar to Fig. 40, and is used in the same manner, with the additional step necessitated by the variable "S_a" entering into the equation. In this case, the result is found plotted upon the left-hand margin of the sheet instead of on the lower edge, as in Fig. 33.

The analysis of the full-load test economy of power and pumping plants is concluded in the above final discussion. Detailed methods have been presented for solving:

(1) Non-condensing power plants.
(2) Surface-condensing power plants (steam-driven auxiliaries).
(3) Surface-condensing power plants (power-driven auxiliaries).
(4) Jet-condensing plants.
(5) Surface-condensing centrifugal or triplex pumping plants.
(6) Non-condensing high duty or direct acting pumping plants.
(7) Surface-condensing high duty or direct acting pumping plants.

Any desired combination of the above may be treated in a similar manner, but it is believed that the examples given will fill practically all ordinary cases.

It should be remembered that in the case of the power plants, the method has been:

(1) To determine the actual steam consumption per hour of the engine and all auxiliaries.
(2) To add these together and multiply the sum by the factor of evaporation.
(3) To divide the product by the equivalent evaporation per barrel of oil, thus obtaining the barrels of oil consumed per hour.
(4) To divide the Kw. hours output per hour by the number of barrels of oil thus found, obtaining the desired result in terms of Kw. hours per barrel of oil.

For engine driven, centrifugal or triplex pumping plants, the hourly consumption of fuel oil in barrels was obtained in exactly the same manner, but this was divided into the foot pounds of work performed per hour, thus obtaining the foot pounds of work accomplished per barrel of oil.

For high duty and direct-acting pumping plants, the same method was pursued with the exception that the fuel consumption was obtained per 1000 lbs. of steam delivered to the engine instead of the hourly consumption.

In conclusion, one more word of caution seems advisable. The principal value of this method of figuring lies in the ability of the engineer to determine by its aid, in advance, the probable economy under test conditions of a proposed plant either for purposes of guaranteeing its economic performance or for some other purpose. Care should be taken to select such conservative values for the different variables as can easily and certainly be obtained, and to further protect the final result by a proper margin of safety.

Upon ten-hour tests, conducted by experts, results of complete plant economy tests have been shown to bear a remarkably close proximity to the results derived by this method of figuring, when the proper values have been assumed in solving the equation, and in cases where discrepancies have appeared, it has been found that the engine, boiler, or aux-

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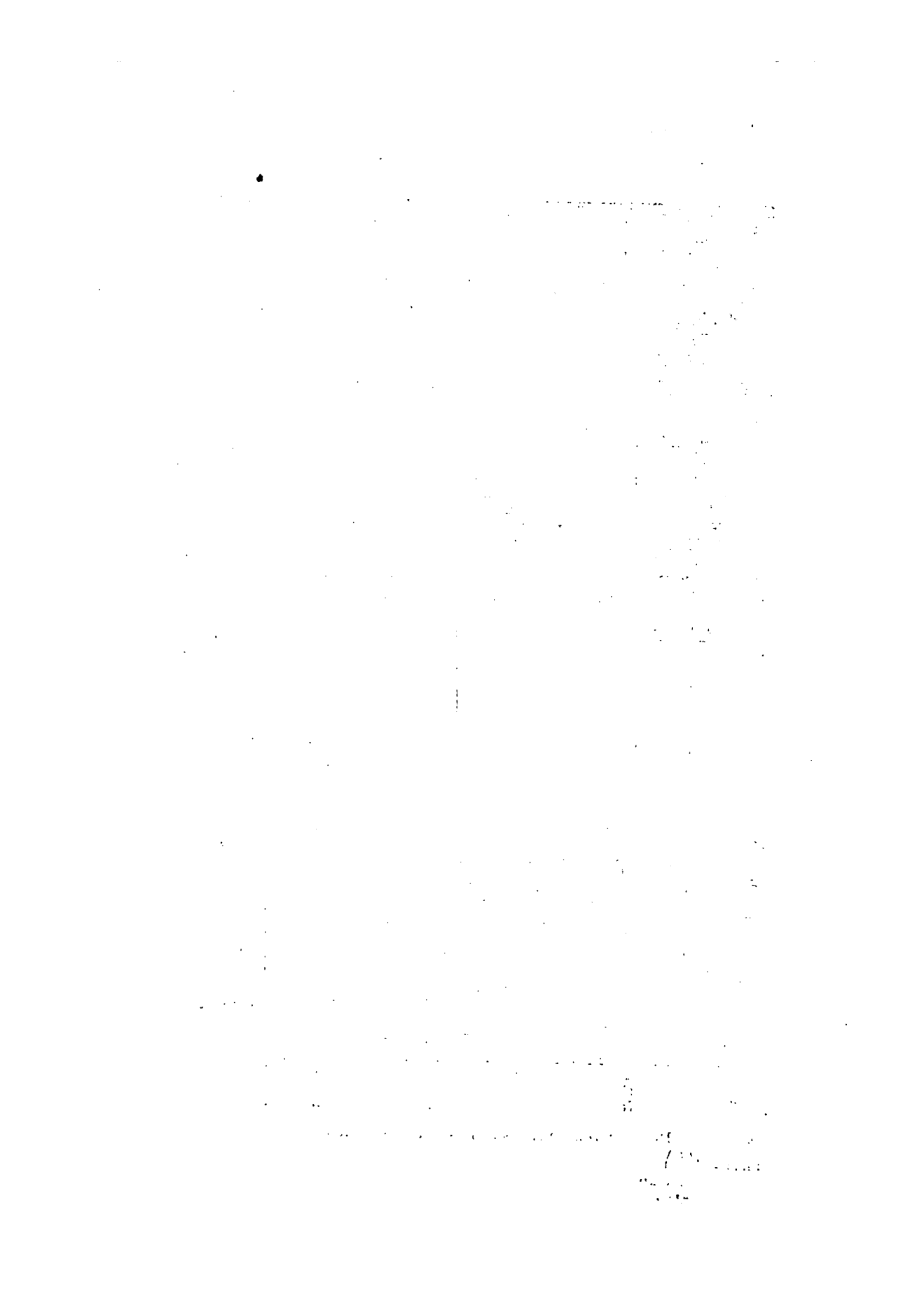
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iliaries have been operating with a correspondingly different economy from that used in arriving at the probable result. However, while we know that a certain engine may be capable of developing one horse-power upon 15 lbs. of steam per hour; that a certain boiler can easily obtain 75% efficiency; that the condensing apparatus can easily maintain a vacuum of 27″ and that 60 lbs. of circulating water per pound of exhaust steam will be sufficient, etc., what assurance have we that all of these conditions will obtain at the same time, and remain so without variation during the entire test?

It is for this reason that a margin of safety becomes necessary; its magnitude depending upon the conservativeness of the variables entering into the calculation.

EXAMPLES

IN order to clear up any points which may have been inadvertently slighted in the foregoing discussions, the following typical examples have been prepared. All illustrations assume high-grade water-tube boilers having specially designed oil furnaces.

EXAMPLE I

Given. — A 200 Kw. non-condensing lighting plant of the belted type, comprising the following apparatus:

Boilers. — Three boilers — two corresponding to the rated capacity of the plant and one for reserve.

Engines. — One unit of the automatic tandem compound piston-valve type, developing its rated capacity with 140 lbs. I. S. P.

Feed and Oil-Pumps. — Duplex direct-acting steam-driven.

Heater. — Enclosed type taking steam from feed-pumps and a portion of main engine exhaust.

Fuel. — California crude oil, 18,850 British thermal units per pound, 336 lbs. per barrel.

Required. — The full-load test economy of the plant in terms of kilowatt hours output at the switchboard per barrel of oil burned.

Solution. — It is at once seen that equation No. 35 exactly meets these conditions, and it is therefore necessary to immediately obtain the proper numerical vaue for "$HE_bE_eE_g$." This, however, involves "E_b," (the boiler efficiency) which in turn involves the size of the boilers used. At the outset, therefore, it is necessary to ascertain the approximate size of the boilers, but this need not be very carefully done as a slight error in the assumed size makes very little difference in the efficiencies selected.

Referring to Fig. 4, the steam consumption of a 300 H. P. engine of the required type is equal to 94.6% × 23, equals 21.758, say 22 lbs. per

I. H. P. hour.   Allowing 10% for the steam consumption of the auxiliaries, the boiler horse-power necessary at full rated load equals, roughly —

$$\frac{300 \times 22}{30} + 10\% = 242 \text{ or about } 250 \text{ B. H. P.}$$

But there are two boilers, according to the proposition, for full-load conditions, so that approximately these will be two 125 H. P. boilers.   We then have,

H	....	18,850
E_b	(from Fig. 2) say	76%
E_e	say	94%
E_g	(from Fig. 13) say	92%
Efficiency of belt drive — say	....	95%
$E_e E_g$	$= .94 \times .92 \times .95$ — say	82%

Hence

$HE_b E_e E_g$	....	11,747
P	$=$ (allowing twenty lbs. drop between boiler and engine cylinder)	
	$140 + 20$	160 lbs.
S_f	....	200 lbs.
PS_f	....	32,000

Hence,

Numerator of equation No. 35 equals (from Fig. 24) about................ 7,118,700

Sufficient steam being available from the engine exhaust, the temperature of the feed will be not less than 200° Fahr., whence,

$$F \quad = \text{(from Fig. 19) } 1.063$$
$$\text{and}$$
$$FS_e = \text{(from Fig. 22) } 23.4.$$

We now have sufficient data for directly applying Fig. 25 from which it will be found that the required value is

$$K_b \quad = 115.$$

EXAMPLE II

Given. — A 1000 Kw. condensing railroad plant — direct connected type — comprising the following apparatus:

Boilers. — Two batteries of two boilers each — three corresponding to the full-rated capacity of the plant, one for reserve.

Engines. — One unit — cross-compound grid-iron valve type — developing rated capacity with 160 lbs. I. S. P. and 24-inch effective vacuum.

Condenser. — Surface type, proportioned for 26½-inch vacuum in shell (allowing drop of 2½″ between condenser shell and engine cylinder), with

circulating water at 70° Fahr.; ratio water to steam 40 to 1; head on circulating pump $h = 25$ ft.

Air and Circulating Pump. — Horizontal, single, steam-driven, direct-acting.

Feed and Oil-Pumps. — Horizontal, duplex, direct-acting steam-driven.

Heater. — Open type, taking steam from feed, air and circulating pumps.

Fuel. — California crude oil 18,500 British thermal units per pound — 336 lbs. per barrel.

Required. — The full-load test economy of the plant in terms of kilowatt hours output at the switchboard per barrel of oil burned.

Solution. — Equation No. 53 will be found to exactly fulfil the conditions of this problem.

To proceed to the solution, from Fig. 9, S_e = say 13¼.

The approximate size of the boilers necessary for full-load conditions is, therefore, $\dfrac{1500 \times 13.25}{30} + 15\% = 762$ — say 750 H. P. According to the problem, however, this capacity is divided among three units, each of which must therefore be ⅓ of 750 or about 250 H. P.

E_b	(from Fig. 2) = say	78%
H	=	18,500
E_e	= say	92%
E_g	(from Fig. 13) = say	95.5%
E_eE_g	= (.92 × .955) say	87.5%
$HE_bE_eE_g$	(from Fig. 21) =	12,626
P	= (160 + 20)	180
S_f	=	150
PS_f	= (180 × 150)	27,000
Numerator (from Fig. 24) =		7,715,000
The percentage of steam to heater (from Fig. 16) =		11.5%

The temperature of steam at 26-inch vacuum is 125.6° Fahr.; allowing a drop of 15° for leakage and for passing through the condenser, the temperature of the air-pump discharge may be taken at 110° Fahr.

Final temperature of feed water (from Fig. 14)		200° Fahr.
Factor of evaporation — "F" — (from Fig. 19)		1.065
FS_e	(from Fig. 22) =	14.1
S_a	=	100
W_h	= (40 × 25)	1000
S_c	=	100
E_c	=	80%
$\dfrac{W_hS_c}{792E_c}$	(from Fig. 26) =	157.83

Recapitulating,

Numerator of equation = ... 7,715,000
FS_e = ... 14.1
S_a = ... 100
$\dfrac{Whs_c}{792E_c}$ = ... 157.83

Applying the above values to Fig. 27, we obtain the kilowatt hours per barrel of oil equal

$$K_h = 187.$$

Example III

Given. — A 3000 Kw. lighting and power plant — direct connected tpye — comprising the following: —

Boilers. — Two batteries of two boilers each.— three corresponding to the full-rated capacity of the plant, one for reserve.

Superheaters. — Boilers provided with combined superheaters capable of superheating steam 120° Fahr. when the plant is operating at its full rated load.

Economizer. — Fuel economizer provided, having one tube per 2 boiler H. P.

Engines. — Three units — cross-compound grid-iron valve type developing full-rated capacity with 160 lbs. I. S. P. and 26-inch effective vacuum.

Condensers. — In three units of the three-pass surface type with primary heating tubes in top of shell — proportioned for 28-inch vacuum in shell — 65° circulating water; ratio of water to steam 66 to 1. Head on circulating pump 15 ft.

Air Pumps. — Vertical triplex suction valveless Edwards type — motor driven.

Feed and Oil-Pumps. — Horizontal, duplex, direct-acting, steam-driven.

Auxiliary Heater. — In addition to primary heater in condenser there is one open type auxiliary heater taking exhaust steam from feed-pumps.

Fuel. — California crude oil — 18,850 British thermal units per pound — 336 lbs. per barrel.

Required. — The full-load test economy of complete plan in terms of kilowatt hours per barrel of oil.

Solution. — This is a three-unit plant, but it will be readily seen that the economy may be the more easily reached by simply treating the plant as a 1000 Kw. one-unit arrangement. Equation No. 68 may be used in this case.

The primary heater in the condenser shell is often used in high vacuum

work as it reheats the air-pump discharge to within a few degrees of the temperature of the steam and is thus of marked benefit.

The variables in equation No. 68 are found as follows:

S_e (from Fig. 9) = ... 11.75

The approximate size of boiler for full load, allowing roughly for auxiliaries, will be $\dfrac{1500 \times 11.75}{30} + 12\frac{1}{2} = 661$ — say 660 H. P. Therefore we have,

E_b	(from Fig. 2) say	82%
H	....	18,850
E_e	=	92%
E_g	=	95.5%
$E_e E_g$	= (.92 × .955)....................................	87.5%

whence

$HE_b E_e E_g$ (from Fig. 21) =	....	13,525
P	= (160 + 20)	180
S_f	=	150
PS_f	....	27,000
Applying the above to Fig. 24, the numerator of the equation is found to be...		8,264,000

From Fig. 17, the percentage of steam used by the feed-pumps and exhausted into the auxiliary heater is found to be $3\frac{1}{4}$%.

The theoretical temperature of steam at 28-inch vacuum is 101.4° Fahr. This temperature will be greatly reduced when coming in contact with the large amount of cold cooling surface in the condenser, but will be reheated by the primary tubes to nearly 100° Fahr. From this data and with an economizer proportioned at 2 boiler H. P. per tube, the final temperature of the feed-water is found to be (from Fig. 15) 200° Fahr.

F	= (from Fig. 20)	1.146
FS_e	= (from Fig. 22)	13.47
W_h	= 990 — say	1000
$E_c E_{cm}$	=	43%
$\dfrac{W_h S_e}{792\,E_e E_g E_c E_{cm}}$	= (from Fig. 28)	40
E_{cm}	=	80%
$\dfrac{S_e}{E_e E_g E_{cm}}$	=	16.8

Recapitulating, we have;

Numerator	=	8,264,000
FS_e	=	13.47
$\dfrac{W_h S_e}{792\,E_e E_g E_c E_{cm}}$	=	40
$\dfrac{S_e}{E_e E_g E_{cm}}$	=	16.8

Applying the above numerical results to Fig. 68, the desired economy equals,

$$K_h = 226.$$

EXAMPLE IV

Given. — A pumping plant, high duty condensing type, consisting of the following machines:

Boilers. — One boiler of sufficient size to operate the entire plant at full rated load.

Engines. — One high duty crank and fly-wheel pumping engine of the triple expansion type, designed for 15,000,000 U. S. gallons of water per twenty-four hours against a total head equivalent to 100 lbs. pressure. Steam pressure 180 lbs., piston speed 240 ft. per minute; r. p. m. 30.

Condenser. — Condenser of the surface type in suction.

Air-Pump. — Horizontal, single, steam-driven, direct-acting.

Feed and Oil-Pumps. — Horizontal, duplex, direct-acting, steam-driven.

Heater. — Closed type, using steam from air and feed-pumps.

Fuel. — California crude oil, 18,500 British thermal units per pound — 336 lbs. per barrel.

Required. — The full load test duty in terms of millions of foot pounds of work performed per barrel of oil burned.

Solution. — This service will require roughly $\dfrac{10,500 \times 231}{4,000}$ — say 600 water H. P. or $\dfrac{600}{.9}$ — about 670 I. H. P.   The size of the boiler will therefore be about $\dfrac{670 \times 14}{30} + 10\%$ — say 350 boiler H. P.

Referring to Fig. 2, it will be seen that a boiler efficiency of 80% may be safely assumed.   From Fig. 37, the duty of the 15,000,000 triple expansion pumping engine under a head of 100 lbs. or 231 ft. is equivalent to $(1.03 \times 136) + 5\%$ which equals 147.09 — say 147 million foot pounds per 1000 lbs. of steam at the main engine.

From the above, we have —

D_p	=	147
H	=	18,500
E_b	=	80%
DHE_b (from Fig. 39) =	....	2,175,600
P	= (180 + 20)	200
S_f	=	200
PS_f	=	40,000
S_a	=	100

From Fig. 16, the per cent steam to auxiliary heater is 12.5 for $Wh = $ 500; 15.75 for $Wh = 1000$; and 19 for $Wh = 1500$, from which it may be

estimated that when $Wh = 0$, the percentage of steam to the heater equals about 9.25%.

With a vacuum of 26″, we can assume the temperature of the air-pump discharge to be 110° Fahr. From this, the final temperature of the feed should be about 198° Fahr. (as shown by Fig. 14), and the factor of evaporation "F" = 1.07 (from Fig. 19).

Substituting the figures determined above in equation No. 105 by means of Fig. 41, the final result is found to be — D = 600 million foot pounds per barrel of oil.

The above actual examples should be sufficient to render clear the method of applying any of the formulæ developed, and of using their accompanying graphs and tables. If the results derived above were intended for use, as guarantees upon a proposed installation, 5% should be subtracted from each one as a "margin of conservativeness," for reasons already explained.

In the following pages variable load economy will be treated under commercial conditions of operation as contrasted with uniform full load economy under test conditions as above set forth.

PART IV

VARIABLE LOAD ECONOMY

UNDER actual operating conditions, it is seldom, if ever, that a steam-power plant approaches a continuous yearly load factor of one hundred per centum. A few cases may be found, such, for instance, as continuously operated pumping plants of certain types, but these examples are so few, comparatively speaking, as to be almost negligible.

It thus develops that in certain cases, the purchaser of a steam-power plant requires a written guarantee of the operating economy of his proposed installation, which figure will be a direct measure of his actual yearly fuel bill, thus aiding, to a great extent, in determining upon the best all-round investment.

The problem to be solved, viz.: the scientific solution of variable load economy thus confronts the various contractors and engineers involved.

It will readily be appreciated that certain types of generators fall off in efficiency more rapidly than others under conditions of fractional and overloads. This fact is clearly illustrated in the curves showing engine economies. Again, different fuels affect boiler efficiencies in different manners under similar conditions, but this treatise has to do primarily with crude petroleum.

LOAD FACTOR

To commence with, let us arrive at a thorough understanding of the term "load factor" as used in this article. "Load factor" may be defined as the ratio of actual power developed to that which would be developed were the units operating at full-rated capacity. But this definition, while generally speaking correct, is somewhat indefinite, and it becomes necessary to subdivide the term "load factor" into two divisions, viz.: "curve load factor," and "absolute load factor."

The "curve load factor" is defined as the ratio between the actual power developed when operating continuously during a given period of time, and the power output under continuous operation at full-rated capacity during the same period of time.

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The "absolute load factor" is the ratio between the actual power developed during a given period of time regardless of whether or not the units are operating continuously, and the power output under continuous operation at full-rated capacity during the same period of time.

To illustrate, — suppose the case of a power plant developing continuously and without variation just one-half of its rated capacity for a period of six months. If, now, the period of time under consideration for which it is desired to determine the load factor is six months also, then both the curve and absolute load factors as defined above, are fifty per centum; but if, on the other hand, the period under consideration is one year or twelve months, the absolute load factor is only twenty-five per centum.

ADDITIONAL VARIABLES

All of the variables entering into the determination of "full load economy" remain to be dwelt with under "variable load economy" and in addition the following:

(1) Magnitude of load factor.

(2) Form of load curves.

(3) Number and size of prime movers installed and reserve units, if any.

(4) Number and size of boilers installed and reserve units, if any.

(5) The "stand-by" losses.

MAGNITUDE OF LOAD FACTOR

The term load factor has been fully described above. There are three general types of plants to be considered with respect to load factor; (1) factory plants, etc., which operate about ten hours per day; (2) office building and railway plants which operate about eighteen hours per day; and (3) railroad and other general service power plants operating twenty-four hours per day. If each of these types of plants should operate continuously for their various periods per day at full-rated capacity the ten-hour plant would have a daily absolute load factor of $\frac{10}{24}$ or about 42%; the eighteen hour plant $\frac{18}{24}$ or 75%; and the twenty-four hour plant $\frac{24}{24}$ or 100%.

FORM OF LOAD CURVE

The form of the load curve should be determined as closely as possible in advance, and should represent the average daily condition of load for the period under consideration. A method has been evolved for the approximate determination of the probable operating economy, independent of the form of the curve, based upon the assumption that the curve representing the total steam consumption of engines and auxiliaries and total

units of power output, is a straight line; also, that the curve showing the total steam produced and total fuel burned under the boilers is another straight line; — while these assumptions are, roughly speaking, correct for normal conditions at extreme under or over loads they are not even approximate. For the present, therefore, this method will be ignored, it being assumed that the form of the load curve is known approximately in each case.

NUMBER AND SIZE OF UNITS

The number and size of generating units must also be known in this method in order to determine the rate at which they are operated during each successive hour. If one of these is a reserve, such information is necessary that the "stand-by" loss described below may be determined. The same statements refer to the boiler installation for the same reasons.

THE "STAND-BY" LOSS

In the commercial operation of a plant under changeable load conditions it is necessary to retain steam pressure on boilers which are otherwise idle in order to provide sufficient reserve capacity for sudden power demands, and to keep up steam pressure during periods when the plant is shut down. Owing to radiation, leakage, and other losses, this requires considerable fuel. Take the case of an office building plant, for instance, operating 18 hours per day. There are here, 6 hours during which no power is being turned out. It is therefore necessary either to keep up sufficient fire to maintain the normal boiler pressure during this period or else to close all dampers and openings and make up the reduced pressure before starting up the plant. Again, during the busy hours of the day — say from 5 to 6 o'clock when all elevators are running and, if in the winter time, nearly all the lights are burning, the power requirement reaches its maximum and all the boilers are operating at a high capacity. But during the light run — say 8 or 9 o'clock on a bright summer morning, only part of the boilers are required, and (if the plant be a large one) probably only one of the generating sets. And, as in the case where the plant is completely shut down, steam pressure must be either maintained or reestablished before the boilers are brought into service. Then there are the losses due to turning over reserve engines preparatory to their being "cut in," and the warming up of idle units, etc. It will be noted that all of this requires fuel while not adding in any degree to the power output. The fuel losses caused by these conditions are known as "stand-by losses" and are common to every plant.

Their magnitude depends upon the load factor, hours of operation per day, number of units, design and construction of plant, kind and effectiveness of non-conducting covering, climatic conditions, etc., and

is therefore a very difficult factor to accurately determine. It will here be assumed, however, that only thoroughly well-designed, up-to-date plants are being considered under ordinary conditions, as they exist on the Pacific Coast, and it is found that under these conditions the stand-by loss every day averages between three and eight per centum of the fuel which would be required to produce the total boiler horse-power hours which are idle per day, provided the boilers were operated at full load. An average of 6% is conservative for any first-class plant. The simplest manner in which to explain this is by reference to an actual example. See Fig. 42, which is a reproduction from an actual daily load curve in a large office building. The plant consists of two units of 75 Kw. each (one of which is used as a reserve only) and one of 150 Kw. rated capacity; and three water-tube boilers of 150 H. P. each (one being for reserve purposes). The generating sets were very high-grade tandem compound self-oiling automatic direct-connected non-condensing piston-valve units. We now have sufficient data for the determination of the probable stand-by loss. The general method is to first establish the points on the load curve where different boilers should be cut in and out, thus dividing the day into several periods during each of which certain boilers are operated. From this is obtained the total boiler capacity idle and the number of hours of such idleness; from which the stand-by loss may be approximated in accordance with the above rule. This involves a knowledge of the boiler efficiencies and engine economies to be obtained which have already been treated separately.

Assuming Fig. 42 to represent the average daily conditions of load during the year, the average curve load factor will be represented by the ratio of the combined shaded areas $A I J K D$ and $E L M H$ and the areas of the combined rectangles $A B C D$ and $E F G H$. In the present case, this is found to be 47.8%.

The yearly absolute load factor will be the ratio of the combined shaded areas $A I K J D$ and $E L M H$ and the rectangle $A B G H$ — which, as will be noted, includes the period when the plant is shut down. This is equal to 35.85%.

But to return to our example in Fig. 42. The exhaust steam from the engines is more than sufficient for furnishing heat for the building so that all the steam generated in the boilers is used by the engine and auxiliaries. Now, in order to determine the proper time for cutting in or out various boilers it is first necessary to predetermine just when different generating units will be required in service. It will be noted that in the example under consideration, the entire day of 24 hours has been divided primarily into five divisions, as follows:

(1) From 12.00 noon to 4.45 P.M.

(2) From 4.45 P.M. to 7.00 P.M.
(3) From 7.00 P.M. to 12.00 midnight.
(4) From 12.00 midnight to 6.00 A.M.
(5) From 6.00 A.M. to 12.00 noon.
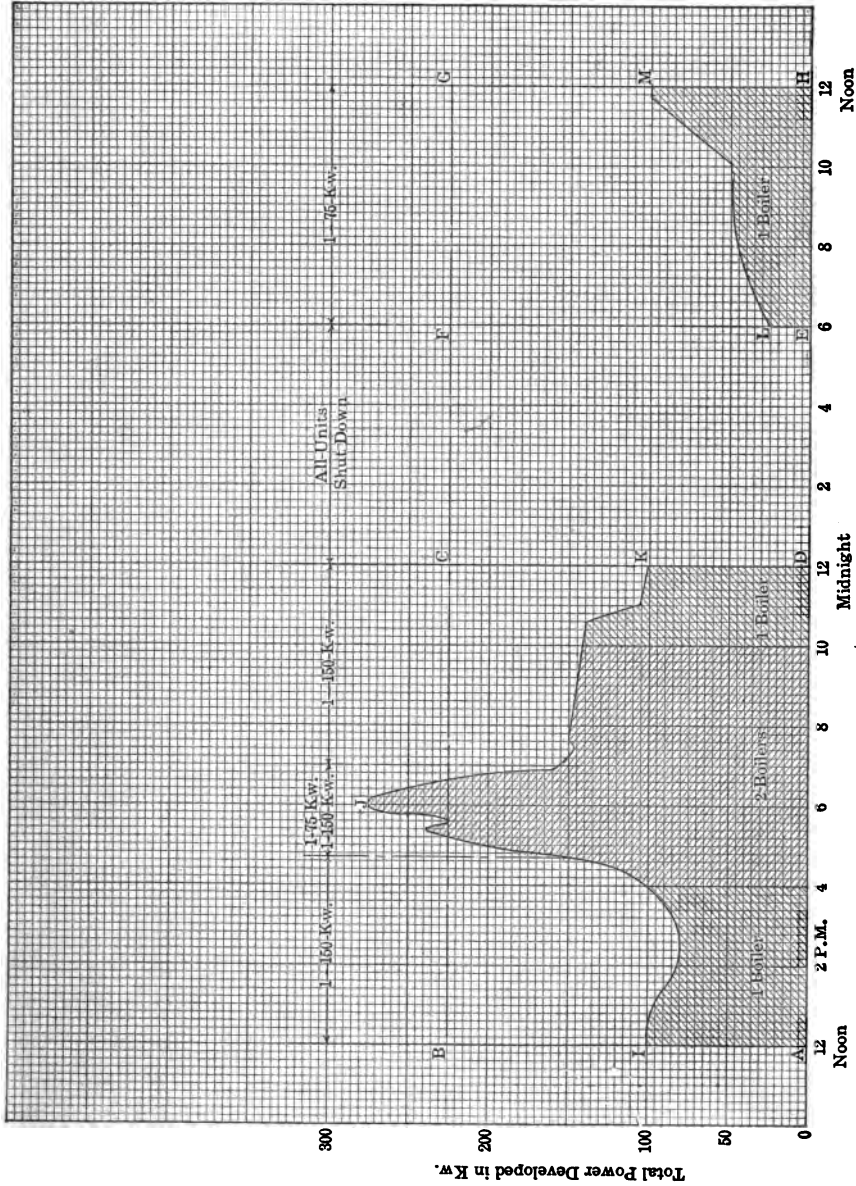
FIG. 42 — OFFICE BUILDING LOAD CURVE

During the first of these periods the maximum output is 150 Kw. and the average about 100 Kw.   One unit is kept in service, viz.: the 150 Kw. machine.

During the second period the load rapidly increases to a maximum of 275 Kw., the average being about 180 Kw.   One 150 Kw. and one 75 Kw. units take care of these conditions operating at a slight overload for about one hour.

The third period is taken care of quite economically by the original 150 Kw. machine, the smaller unit having been shut down at 7.00 P.M.

From midnight until 6 o'clock in the morning the entire plant is out of service.

The last period — that during the morning hours — is well taken care of by one 75 Kw. set.

Knowing now the size and type of the generating units in operation at any time during the day, and the load under which they operate at that time, the determination of the amount of steam required and therefore the number of boilers necessary at that moment is rendered easy.   Say, for instance, the plant is started up at 6.00 A.M. with one boiler in service. The steam requirements are easily within the capacity of this boiler until the load suddenly increases late in the afternoon.   At 4.00 P.M 100 Kw. is being generated, the unit being operated at about two-thirds of its rated load at that instant.   Assuming the steam consumption to be 25 lbs. per I. H. P. per hour and a combined engine and generator efficiency of about 90%, there will at that moment be required steam at the rate of approximately 4300 lbs. per hour.   This allows about 15% for auxiliaries, etc., and is equivalent to 145 boiler horse-power — roughly speaking. ˙This is a very small load for a good water-tube boiler of 150 H. P. capacity burning oil fuel, as is shown by Fig. 2, but in the example under consideration, the load increases very suddenly after 4.00 P.M., so that it is necessary to cut in the second boiler at that time.

At 6 o'clock the load has reached its maximum of 275 Kw. and there is then required for the entire plant steam at the rate of about 12,000 lbs. per hour or 400 boiler horse-power.   This condition, however, is only instantaneous, the demand increasing and decreasing very rapidly before and after this time.   Two 150 H. P. boilers are therefore easily capable of caring for these conditions until about 10.00 P.M. when one boiler is cut out, leaving the remaining one in operation until the plant is shut down at midnight.

It is therefore easily seen that the day is further divided into four periods of operation in the boiler-room which may be summarized as follows:

(1) From 6.00 A.M. to 4.00 P.M., one boiler used.

(2) From 4.00 P.M. to 10.00 P.M., two boilers used.

(3) From 10.00 P.M. to 12 midnight, one boiler used.

(4) From 12 midnight to 6.00 A.M., no boilers used.

From this is developed the number of boiler horse-power idle during the day, neglecting the reserve boiler which is always idle and which is not considered for the reason that it is not kept under steam and therefore uses no fuel.

There is one boiler idle, but under steam, from 6.00 A.M. (when the plant is started up) until 4.00 P.M. — or 10 hours; there is one boiler idle from 10 P.M. to 12 midnight or 2 hours; and there are two boilers idle from 12.00 midnight to 6.00 A.M. — or 6 hours each. This is equivalent to one 150 H. P. boiler being idle $10 + 2 + (2 \times 6) = 24$ hours. Now if this one 150 H. P. boiler were actually operating at full load for 24 hours there would be developed $150 \times 24 = 3600$ boiler H. P. hours. Assuming fuel of a calorific value of 18,500 B. T. U. per pound and a boiler efficiency of 75%, there would be required; $\dfrac{3600 \times 34\frac{1}{2} \times 966 \times 4}{336 \times 18{,}500 \times 3} = 26$ barrels of oil per day — approximately. But the actual stand-by loss is about 6% of this or $6\% \times 26 = 1.56$ barrels per day.

It should be distinctly understood that the above illustration is simply an example designed to make clear the definition of the term "stand-by" loss as used in this article and the general method of arriving at its numerical value, and does not pretend to extreme accuracy of result. As a matter of fact, however, the above determination agrees in a remarkably close degree to the actual stand-by loss in the office building plant in question when the load conditions as set forth on the accompanying load curve were obtained.

With the above explanation in mind, the application of this principle to the variable load formulæ to follow will be readily understood.

METHOD OF CALCULATING VARIABLE LOAD ECONOMY

The general process in solving for variable load economy is as follows:

(1) Establish on the load curve several points dividing the day into periods during each of which certain engines and boilers are in operation in a similar manner as in the above-described determination, of the "stand-by losses."

(2) Knowing the average load at which the engines and boilers are required to operate, determine by means of one of the formulæ for full load economy above developed, the corresponding fuel consumption of the plant for each successive hour of operation.

(3) Add the results obtained, together, thus getting the apparent fuel consumption of the plant per day due to the power actually turned out.

(4) Apply the proper correction as explained below for determining the actual fuel consumption due to turning out power.

(5) Determine the "stand-by" losses as explained above and add the fuel thus formed to that used in turning out power.

(6) Divide the kilowatt hours output per day by the total barrels of oil consumed, resulting in the plant economy expressed in terms of kilowatt hours per barrel of oil.

COEFFICIENT FOR CORRECTION OF RADIATION AND LEAKAGE

All of these items should be self-explanatory with the exception, perhaps, of item 4.   It will be remembered that in developing the formulæ for full-test load economy, the steam consumptions of the oil burners, oil-pumps, and radiation and leakage were assumed at 3%, 1% and 3% respectively, making a total of 7% of the entire steam used by the plant.   But this was at full-rated load.   At a small fractional load the actual steam lost by radiation and leakage would be about the same as at full load, so that if 3% of the steam consumption at the small load were taken, the result would be insufficient.   This per centage should be referred to the steam consumption of the plant at full-rated load.

One per cent of the actual steam consumption for the oil-pumps is sufficient in any case, and 3% suffices for the oil burners.   It is therefore necessary either to develop new formulæ for the hourly fuel consumption under variable load or to use those above demonstrated in connection with a proper coefficient of correction depending on the load factor.

Assuming the load factor were twenty per centum, and the relative economy were the same as at full load, there would be wasted by radiation and leakage 5 × 3 or 15% of the steam actually generated in place of 3% as at full-rated load.   But the actual steam economy of the plant per kilowatt turned out would be poorer at 20% than at 100% load factor, so that this 15% requires further modification, in accordance with the relative economy of the plant at fractional loads. · Inasmuch as the hourly steam consumption of the auxiliaries is nearly a function of that of the main engine, the relative economy of the latter may be taken as a basis for figuring the coefficient of correction for radiation and leakage.

Accordingly, Fig. 43 may be used in applying proper corrections to the daily fuel oil consumption of any given plant under variable load conditions.   It should be remembered that this coefficient will vary not only with the load factor but also with the ratio of falling off in economy due to fractional load.

Four curves are therefore plotted: (1) for plants having any compound non-condensing engines; (2) for plants having any single cylinder non-condensing or compound-condensing engines; (3) for plants having grid-iron valve engines operating with superheated steam, and finally

(4) for a theoretical plant having engines, the steam consumption per H. P. hour of which was a constant for all loads. The product of the economy of a complete plant at a fractional load as obtained from the formulæ given above for full-load economy and the coefficient for the same load will give the final economy corrected for radiation and leakage as

FIG. 43. — SHOWING "COEFFICIENTS FOR RADIATION AND LEAKAGE CORRECTION"

explained. It will be noted that the less the engine economy falls off at small loads the smaller the coefficient and hence the smaller the result.

Having defined the additional variables involved, a formula will be developed for variable load economy. This will necessitate the following additional notation:

K_{hv} = the final net Kw. hours per barrel of oil for variable load.

K_{hm} = mean hourly value of K_h for the actual number of hours per day under operation.

O = number of hours operation per day.

L = curve load factor.

C = coefficient for correction for radiation and leakage.

S = number of boiler horse-power hours idle per day and producing "stand-by" losses.

The symbol "K_h" will be used to designate the economy of the plant in terms of kilowatt hours per barrel of oil for each separate hour during the day, neglecting "stand-by" losses and the correction for radiation and leakage — in other words, "K_h" will be used in practically the same sense as it was used under full-load economy.

Letting Σ denote the process of summation the mean hourly kilowatt hours per barrel of oil, ignoring the correction for radiation and leakage equals

$$K_{hm} = \frac{\Sigma K_h}{O} \qquad \qquad \text{(106)}$$

Applying the coefficient for correction for radiation and leakage, we have the actual average kilowatt hours per barrel of oil during the period of daily operation due to turning out power equals

$$CK_{hm} = \frac{C\Sigma K_h}{O} \qquad \qquad \text{(107)}$$

From this the actual fuel oil in barrels consumed per day, due to turning out power will equal

$$\frac{KOL}{CK_{hm}} = \frac{KO^2L}{C\Sigma K_h} \cdot \qquad \qquad \text{(108)}$$

In addition to the fuel, represented in equation (108), which is that consumed due to actually turning out power, there is required the fuel due to keeping up steam, etc., the "stand-by" losses. These may be estimated thus:

The equivalent pounds of steam lost per day from and at 212° Fahr. by radiation and leakage due to idle engines and boilers, equal:

$$.06 \times 34\tfrac{1}{2} \times S \qquad \qquad \text{(109)}$$

Dividing this by the equivalent evaporation in the boilers per barrel of oil we have the barrels of oil lost per day from this cause, which is

$$\frac{.06 \times 34\frac{1}{2} \times 966\,S}{336\,HE_b} = \frac{5.95\,S}{HE_b} \qquad \cdots \qquad (110)$$

Adding (108) and (110) we have the total fuel consumption of the plant in barrels per day:

$$\frac{KO^2L}{C\Sigma K_h} + \frac{5.95\,S}{HE_b} \qquad \cdots \qquad (111)$$

Dividing the kilowatt hours developed per 24 hours (KOL) by the barrels of oil burned in that time (equation 111), the final result in terms of kilowatt hours per barrel of oil will be represented by

$$K_{hv} = \frac{KOL}{\dfrac{KO^2L}{C\Sigma K_h} + \dfrac{5.95\,S}{HE_b}} \qquad \cdots \qquad (112)$$

Equation (112) is correct for any type of a steam electric plant whatsoever. In applying, it is necessary only to substitute for K_h the proper results as derived from the equation representing the full-load economy of the particular kind of plant under consideration.

TEMPERATURE OF THE FEED-WATER — VARIABLE LOAD

In solving for the numerical value of K_{hv}, the values of K_h are obtained from the full-load economy tables and graphs, all of which may be directly applied with the exception of Figs. 14 and 15, giving feed-water temperatures. At a fractional load the temperature of the feed will be higher than at the full-rated capacity of the plant by reason of the fact that at underloads the steam consumption of the auxiliaries is not so well sustained as that of the main units, thus producing a larger per centage of exhaust steam for heating purposes at fractional loads. The plant as a whole, may, however, be operating at a small fractional load factor while any main unit is well loaded. This occurs in a two-unit plant when one machine is out of service, for instance. In the latter case, the plant may be operating with practically the same temperature of feed-water as were the load factor 100%. In other words, it is the load upon each individual unit and upon the auxiliaries which varies the feed-water temperature and not the plant load factor.

Figure 44 will be found useful in determining the relation between the temperature of the feed-water when the plant is developing its full-rated load and when the units are partially loaded.

Having obtained the temperature of the feed-water from Fig. 44, and the values of K_h for each hour, the latter results may be added together, thus obtaining the required value of ΣK_h. C is obtained from Fig. 43 as already explained. Figure 45 shows graphically values of KOL and KO^2L for various values of K, O, and L. The size of the plant or genera-
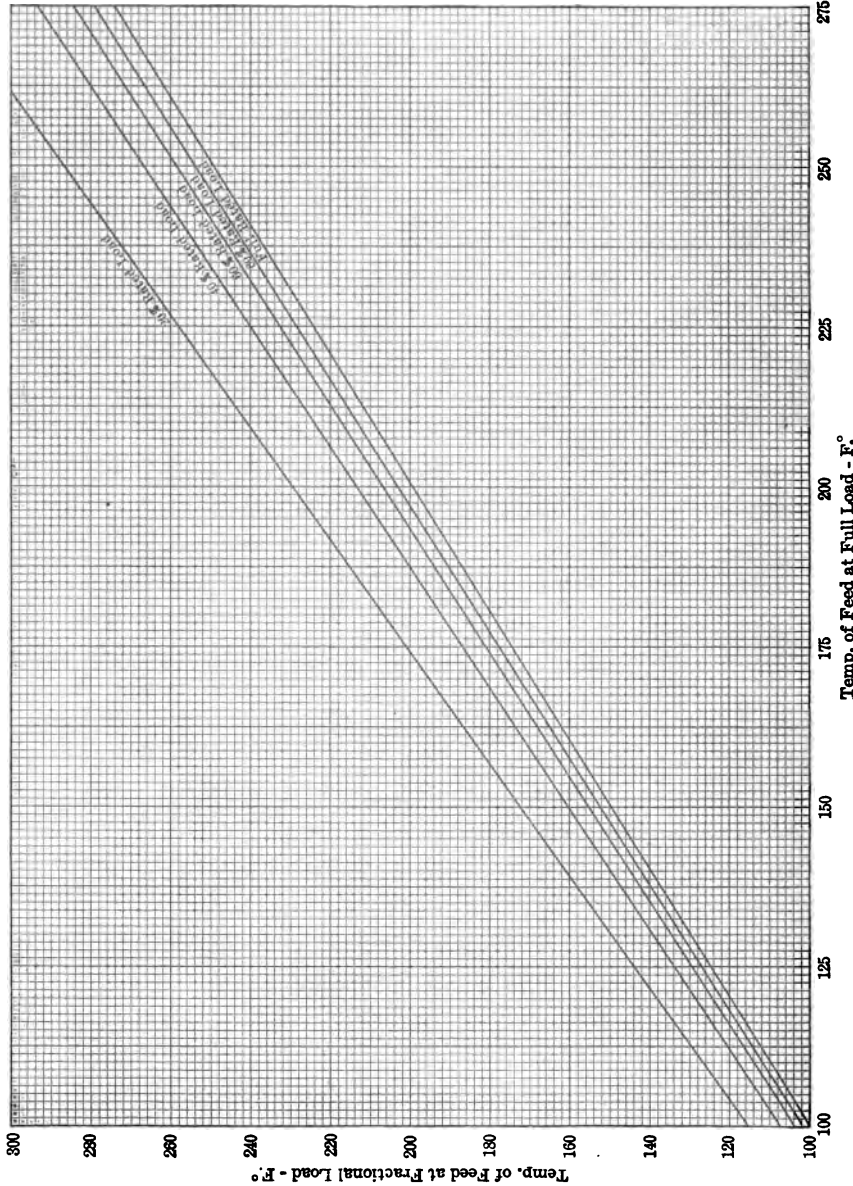
FIG. 44. — SHOWING THE RELATION BETWEEN FEED-WATER TEMPERATURES AT FULL AND FRACTIONAL LOADS

ting unit K is plotted on the lower horizontal margin, the values of KOL upon the upper and KO^2L on the left margin. The dotted lines are used only in determining values of KO^2L. This is done by moving horizontally
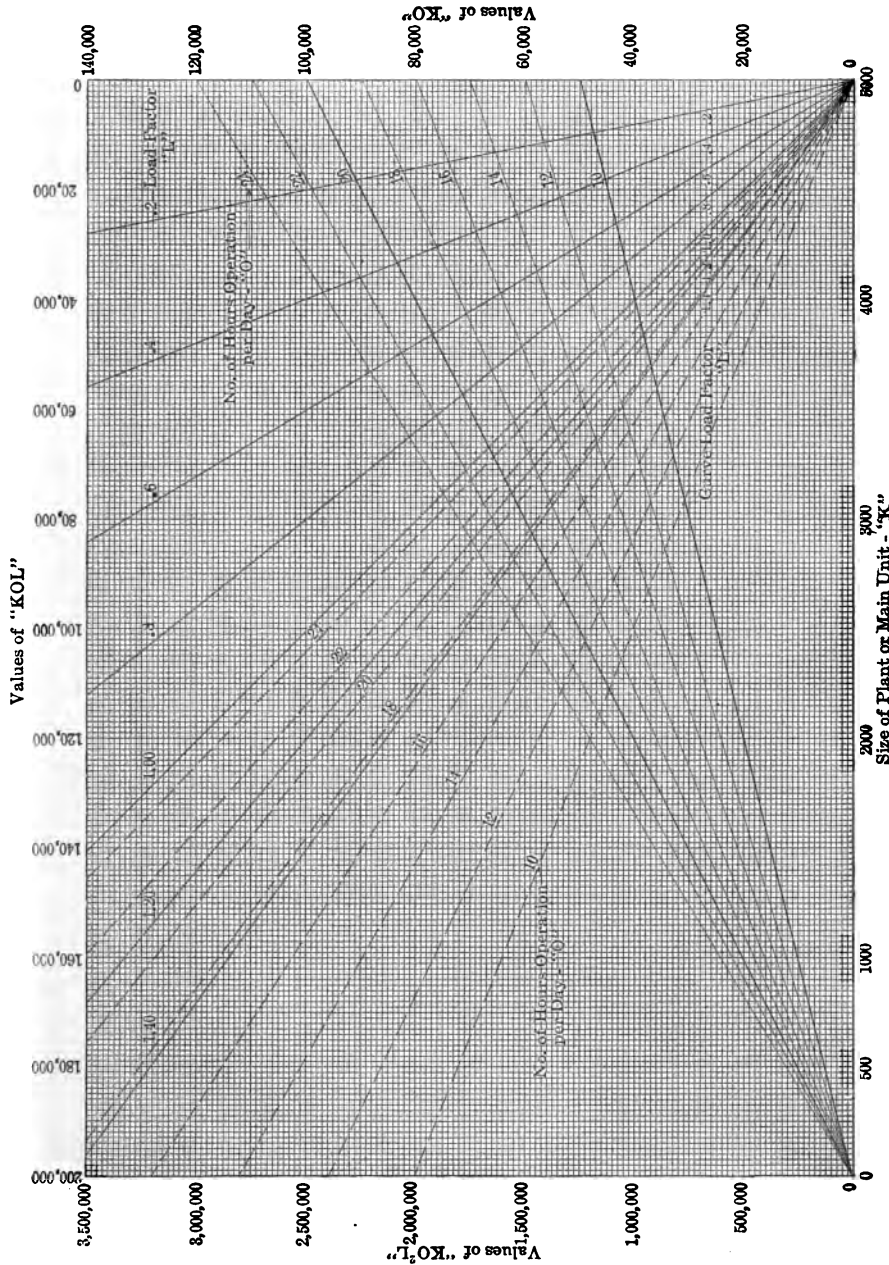
Fig. 45. — Values of "KOL" and "KO^2L"

from the intersection of the vertical line denoting the proper capacity of the plant and the full line denoting the proper number of hours of operation per day, until reaching the full line showing the required value of L, thence vertically to the dotted line showing the same values of O, thence horizontally to the required value of KO^2L at the left of the chart.
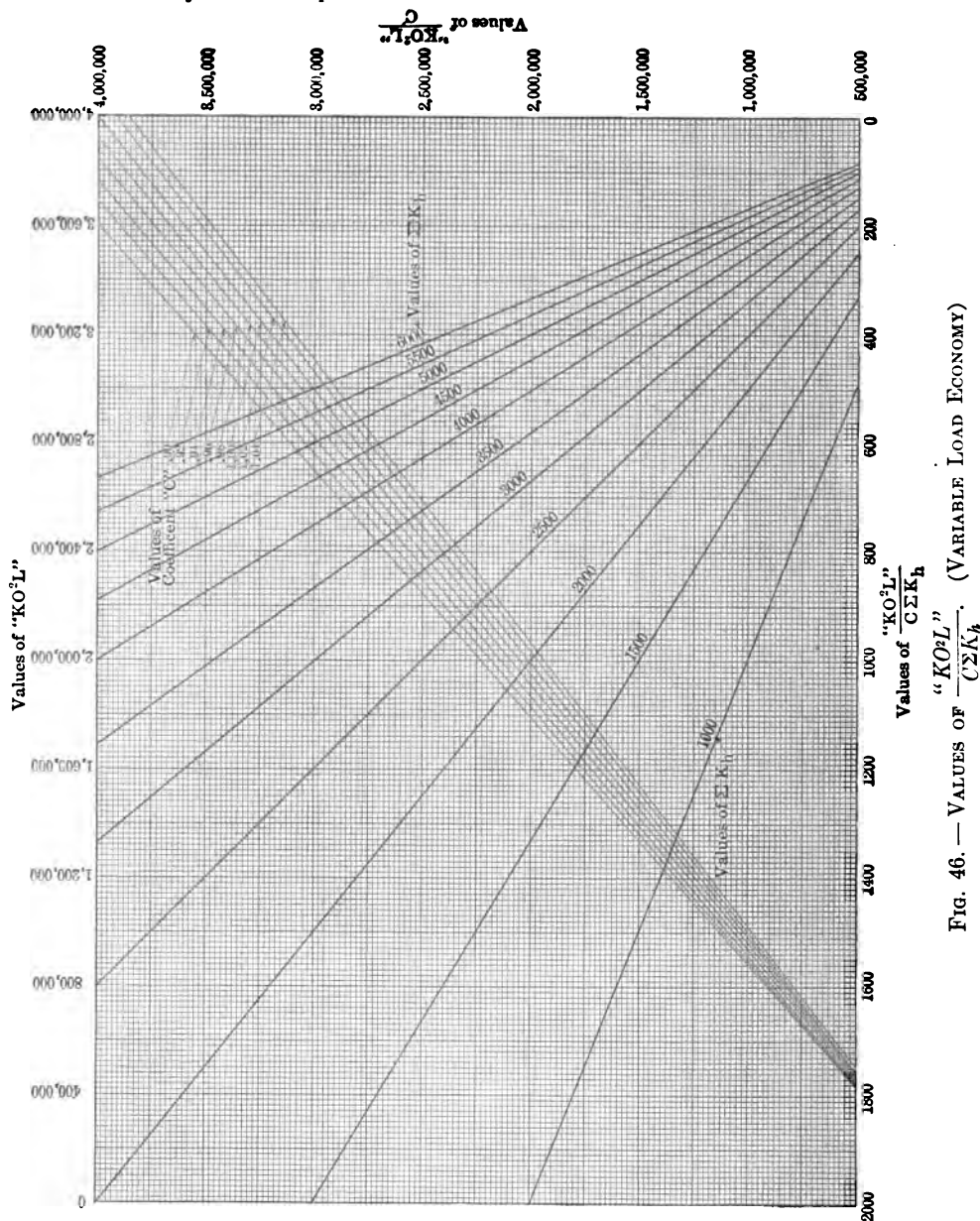
Fig. 46. — Values of $\dfrac{``KO^2L"}{C\Sigma K_h}$. (Variable Load Economy)

Figure 46 represents $\frac{KO^2L}{C\Sigma K_h}$ for various values of "KO^2L," "C" and "ΣK_h." The numerical result is obtained by selecting the proper value of "KO^2L" upon the upper scale, moving downward to "C," horizontally to "ΣK_h" and downward to the lower horizontal scale where the required values will be found.
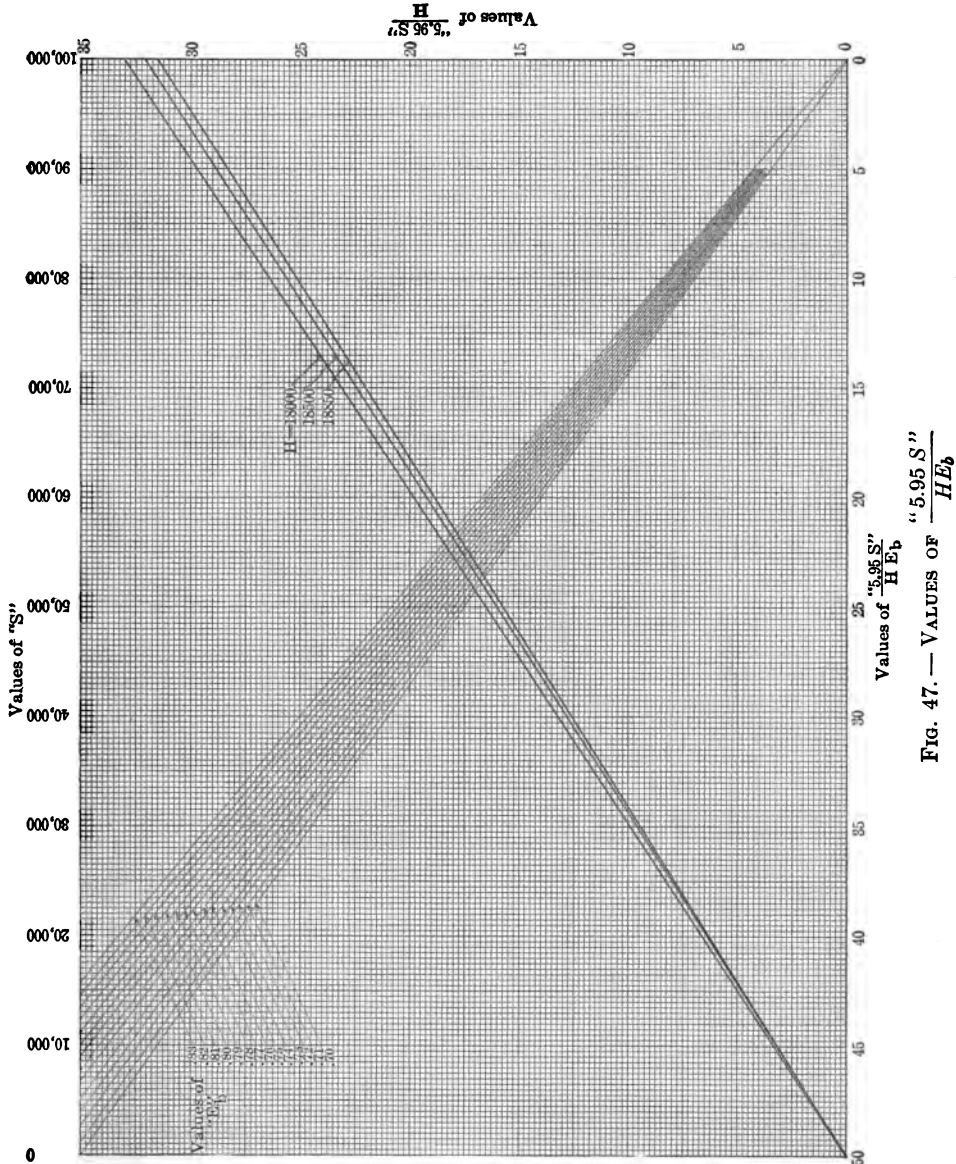
FIG. 47. — VALUES OF $\frac{\text{"5.95 }S\text{"}}{HE_b}$

Figure 47 sets forth values of $\dfrac{5.95\,S}{HE_b}$ for required values of "S," "H" and "E_b." "S" must be obtained from the load curve as already explained. In using Fig. 47, commence with "S" on the upper scale, move downward to "H," horizontally to "E_b" and vertically to the required lower scale.

Having obtained numerical values for the above expressions, Fig. 48 may be used in applying equation (112). By locating $\dfrac{KO^2L}{C\Sigma K_b}$ upon the right-hand margin moving horizontally to one of the parallel lines denoting values for "$\dfrac{595.S}{HE_b}$" vertically to one of the curved lines and finally straight out to the left-hand margin, the result may be directly read in terms of kilowatt hours per barrel of oil.

EXAMPLE

It would seem appropriate, before concluding, to illustrate more fully the process of determining the economy of a power plant under variable load conditions by means of an actual example, working out each step in detail. Accordingly, Fig. 49 is presented, which represents a hypothetical load curve from an electric railroad plant. The problem is as follows:

Given. — An electric railroad plant of 5000 Kw. capacity, operating under a curve load factor of about 55%. The average conditions of load to be in accordance with the plotted load curve upon Fig. 49. The plant contains the following apparatus:

Boilers. — Six high-grade water-tube boilers built for 200 lbs. working pressure, each boiler being rated at 600 H. P. Special furnaces provided for burning crude oil. Five boilers for regular service, one for reserve.

Superheaters. — Each boiler provided with superheaters for 125° Fahr. superheat.

Engines. — Two main units of the cross-compound grid-iron valve type, each developing its rated capacity of 2500 Kw. with 165 lbs. I. S. P., at about 25% cut-off. Both engines and generators good for continuous load 25% above rated, and a load for 2 hours of 50% above rating. Engines operate condensing with 26-inch vacuum.

Condensers. — One for each engine, of the rectangular surface type, proportioned for 28-inch vacuum with 65° circulating water at a ratio of 70 to 1. Head on circulating pump 14¼ ft.

Air-Pumps. — Steam-driven twin-vertical crank and fly-wheel suction valveless Edwards type, one for each condenser.

Circulating Pumps. — Of the centrifugal direct-connected engine-driven type, one for each condenser.

Feed-Pumps. — Horizontal, duplex, steam-driven, direct-acting.

Primary Heater. — Heating tubes provided in top of condenser shell through which air-pump discharge is pumped.

Auxiliary Heater. — Of the open type obtaining steam from exhaust of feed, air, and circulating pumps.
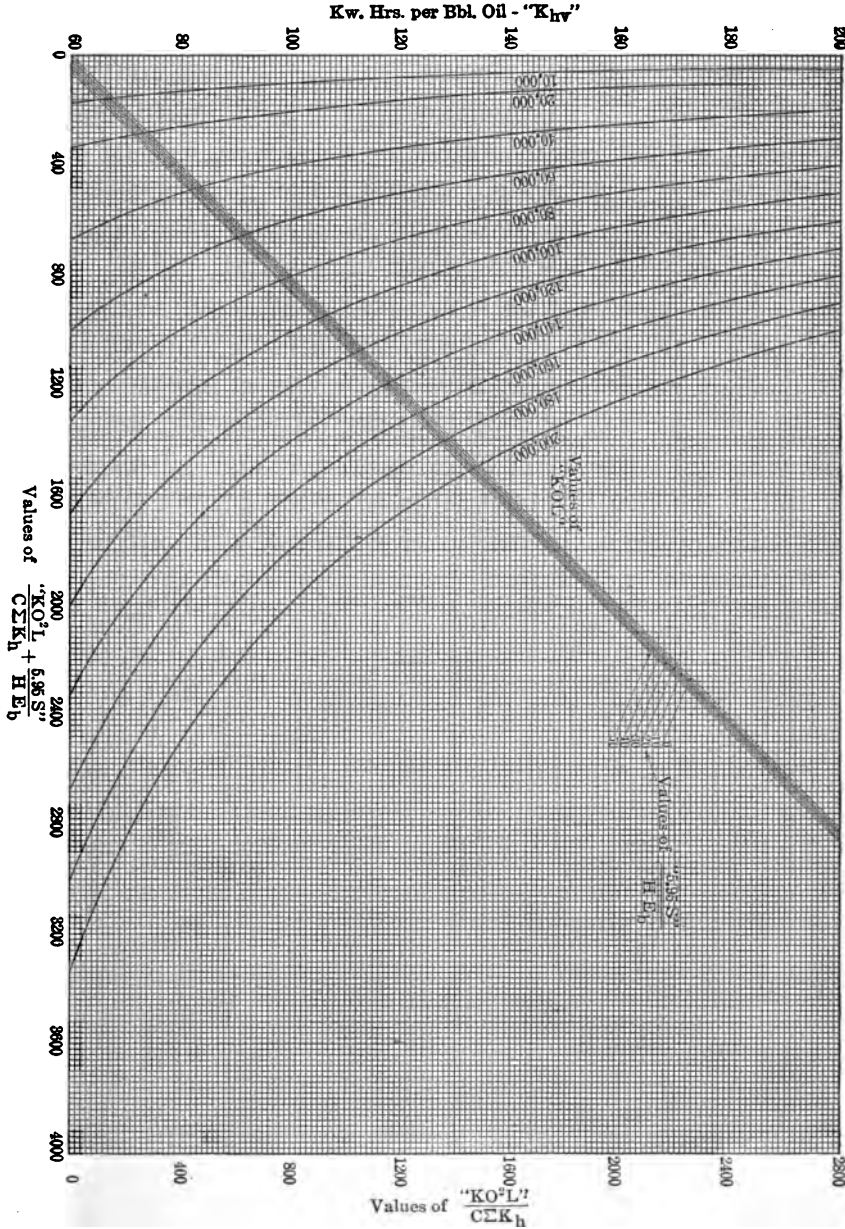
FIG. 48. — GRAPHICAL REPRESENTATION OF EQUATION No. 112. (VARIABLE LOAD ECONOMY)

$$``K_{hv} = \dfrac{KOL}{\dfrac{KO^2L}{C\Sigma K_h} + \dfrac{5.95\,S}{HE_b}}"$$

Fuel. — California crude oil, 18,850 British thermal units per pound, 336 lbs. per barrel.

Required. — The variable load economy including all "stand-by" losses in terms of net kilowatt hours produced at the switchboard per

Fig. 49. — Railroad Plant Load Curve. Illustrating Method of Figuring Variable Load Economy

(Curve Load Factor = 55%)
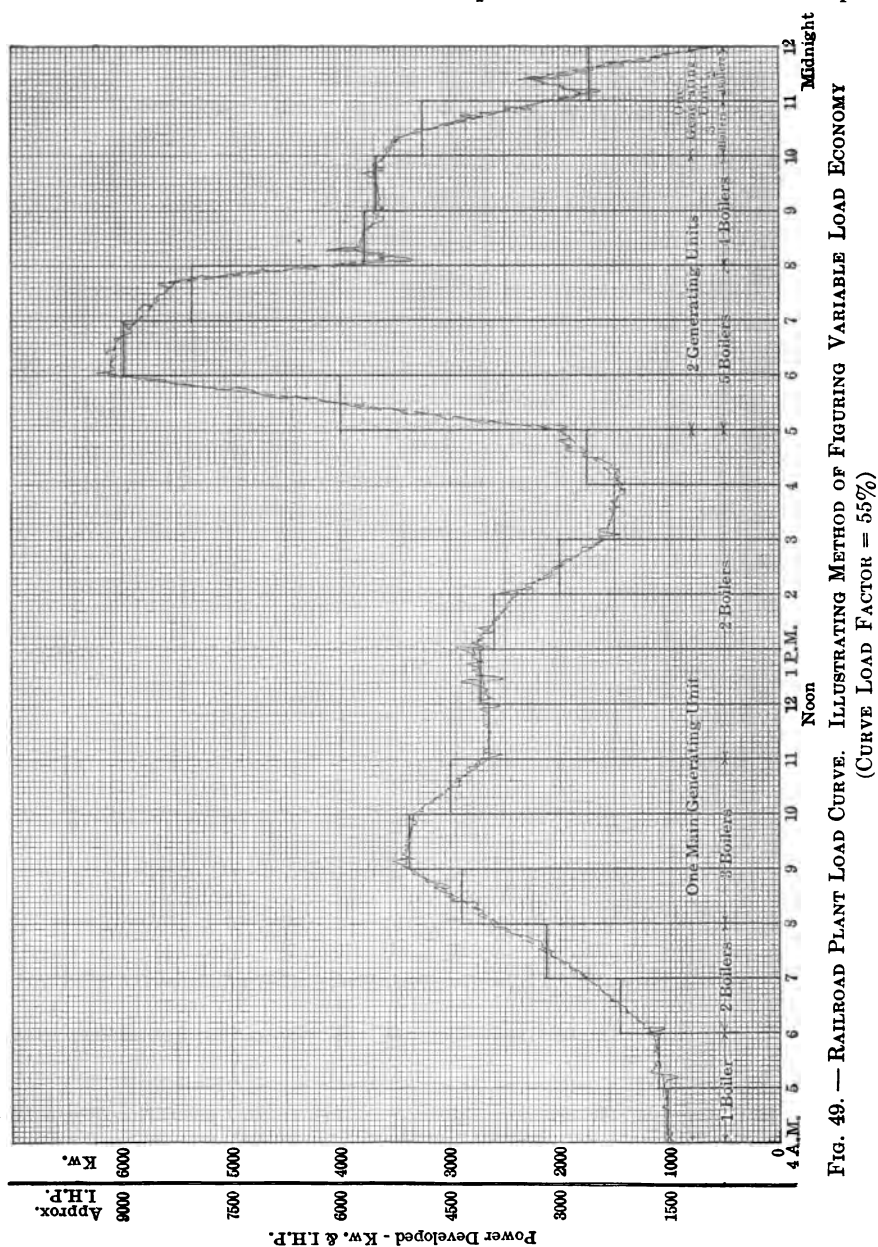
barrel of oil burned under the boilers, the curve load factor of 55% being taken into consideration.

Solution. — It will be found profitable in a problem of this kind to commence by taking the necessary measures to determine the value of "K_h" for each successive hour of operation. But in order to economize work and maintain a systematic method of procedure, a table should be constructed as follows:

Hour	Approx. Average Load I. H. P.	No. of Units Operated	Per Cent. Rated Capacity on Units in Operation	Lbs. Steam per I. H. P. hr. "S_e"	Average Total Steam Required (Approximate)
4 to 5 A.M.	1500	1	40	11.8	20,355
5 to 6	1650	1	44	11.8	21,240
6 to 7	2100	1	56	11.7	28,255
7 to 8	3150	1	84	11.6	42,020
8 to 9	4350	1	116	11.8	59,030
9 to 10	5000	1	133	12.2	70,150
10 to 11	4500	1	126	11.9	61,580
11 to 12 noon	4000	1	107	11.7	53,820
12 to 1 P.M.	4100	1	110	11.7	55,165
1 to 2	3850	1	103	11.6	51,360
2 to 3	3000	1	80	11.6	39,920
3 to 4	2250	1	60	11.7	26,325
4 to 5	2600	1	70	11.7	34,985
5 to 6	6000	2	80	11.6	79,840
6 to 7	9000	2	120	11.9	123,165
7 to 8	8000	2	107	11.7	107,640
8 to 9	5650	2	75	11.6	75,370
9 to 10	5450	2	73	11.6	72,700
10 to 11	4850	1	129	12.0	66,930
11 to 12 midn't	2600	1	70	11.7	34,985
12 to 1 A.M.					
1 to 2					
2 to 3					
3 to 4					

The hours of operation in the first column are taken from the load curve. It will be noted that the plant operates 20 hours per day, thus having 4 hours of complete stand-by daily.

The approximate load in I. H. P. set forth in column No. 2 is obtained as follows:

A line is drawn on the load curve which represents an average load and eliminates the sudden power fluctuations and "railroads wings" of short duration. See the light dotted line upon Fig. 49. The line thus obtained is again modified by dividing it into average hourly powers. In other words, a certain output in power units is determined upon for each

hour which represents the average power developed during that hour. In view of the fact that extreme accuracy is of little consequence in this column (a slight error not being discernible in the final result), the I. H. P. has been determined by multiplying the kilowatt capacity as shown on the load curve by 1.5.

Column No. 3 shows the number of generating units in operation, each being of 2500 Kw. capacity according to the problem. This is also indicated upon the load curve.

Column No. 4 is self-explanatory. It shows the rate at which each unit is operating and is on an I. H. P. basis.

Column No. 5 is obtained from Fig. 9, and assumes 50° Fahr. superheat at the throttle, the remainder being lost in passing through the piping, separators, etc. The results in this column are about midway between saturated steam and 100° Fahr. superheated steam economies.

Column No. 6 is obtained by taking the product of columns No. 2 and No. 5 and multiplying the product by 1.15 to allow roughly for auxiliaries.

Having obtained the above figures, the table may be extended as follows:

Hour	Approx. B. H. P.	No. of Boilers Used	Per Cent. Rated Capacity Boilers in Operation	Boiler Efficiency (Per Cent.)	No. of Boilers Idle	$E_e E_g$ (Per Cent.)
4 to 5 A. M.	679	1	113	78	4	85
5 to 6	708	1	118	77	4	86
6 to 7	942	2	79	75	3	87
7 to 8	1400	2	117	77	3	89
8 to 9	1968	3	109	78	2	92
9 to 10	2338	3	130	76	2	92
10 to 11	2053	3	114	78	2	93
11 to 12 noon	1794	2	150	75	3	92
12 to 1 P.M.	1839	2	153	75	3	92
1 to 2	1712	2	143	76	3	92
2 to 3	1331	2	111	78	3	89
3 to 4	877	2	73	75	3	87
4 to 5	1166	2	97	78	3	88
5 to 6	2661	5	89	77	0	88
6 to 7	4106	5	137	76	0	93
7 to 8	3588	5	120	77	0	92
8 to 9	2512	4	105	78	1	88
9 to 10	2423	4	101	78	1	88
10 to 11	2231	3	124	77	2	93
11 to 12 m'd't	1166	2	97	78	3	87
12 to 1 A.M.					5	
1 to 2					5	
2 to 3					5	
3 to 4					5	

Column No. 2 is obtained by dividing the total steam required by 30.

The number of boilers in operation each hour are shown on column No. 3. See the load curve, also. It is often necessary to place more boilers in operation in a given hour than are necessary so as to be prepared for a sudden demand for steam.

Column No. 4 assumes that the load is divided equally between all boilers in operation.

It will be noted that the boiler efficiencies in column No. 5 are much, below the values given in Fig. 2, but it should be remembered that the latter are based upon steady load test conditions, *i.e.*, the boiler is assumed to operate continuously at the same rate so that air spaces in the furnace, etc., may be carefully adjusted and the best results obtained. This is not possible on a constantly changing load so that a larger excess air supply becomes necessary with a correspondingly reduced boiler efficiency.

The sum of the figures in column No. 6 or 65 will represent the number of "boiler hours" of stand-by. Each boiler being of 600 H. P. capacity, there will be 600×65 or 39,000 boiler H. P. hours of stand-by to account for in the plant under consideration. Thirty-nine thousand is therefore the numerical value for "S."

Values of "$E_e E_g$" in column No. 7 are assumed for each load.

With the above data in hand, a third and final addition to the table should be constructed in accordance with the following:

Hour	$HE_b E_e E_g$	PS_f	Numerator Equation No. 53	Temperature of Feed F°	
				Full Load	Actual Load
4 to 5 A.M.	12,498	18,000	7,750,000	172°	184°
5 to 6	12,483	18,000	7,740,000	172	182
6 to 7	12,300	18,000	7,625,000	172	181
7 to 8	12,918	18,000	8,010,000	172	174
8 to 9	13,527	18,000	8,395,000	172	174
9 to 10	13,180	18,000	8,172,000	172	175
10 to 11	13,674	18,000	8,482,000	172	174
11 to 12 noon	13,007	18,000	8,065,000	172	172
12 to 1 P.M.	13,007	18,000	8,065,000	172	172
1 to 2	13,180	18,000	8,172,000	172	172
2 to 3	13,086	18,000	8,112,000	172	175
3 to 4	12,300	18,000	7,625,000	172	179
4 to 5	12,939	18,000	8,024,000	172	177
5 to 6	12,773	18,000	7,920,000	172	175
6 to 7	13,323	18,000	8,266,000	172	174
7 to 8	13,353	18,000	8,280,000	172	172
8 to 9	12,939	18,000	8,024,000	172	175
9 to 10	12,939	18,000	8,024,000	172	177
10 to 11	13,498	18,000	8,375,000	172	174
11 to 12 m'dn't	12,792	18,000	7,930,000	172	178

Factor of Evaporation	FS_e	$\dfrac{WhS_c}{792E_c}$	S_a	K_h	Hour
1.163	13.72	91.83	40	202	4 to 5 A.M.
1.165	13.75	91.83	40	201	5 to 6
1.166	13.64	91.83	40	201	6 to 7
1.173	13.61	91.83	40	211	7 to 8
1.173	13.84	91.83	40	218	8 to 9
1.172	14.30	91.83	40	204	9 to 10
1.173	13.96	91.83	40	217	10 to 11
1.176	13.76	91.83	40	210	11 to 12 noon
1.176	13.76	91.83	40	210	12 to 1 P.M.
1.176	13.64	91.83	40	215	1 to 2
1.172	13.60	91.83	40	213	2 to 3
1.168	13.67	91.83	40	201	3 to 4
1.170	13.69	91.83	40	210	4 to 5
1.173	13.61	91.83	40	208	5 to 6
1.173	13.96	91.83	40	212	6 to 7
1.176	13.76	91.83	40	217	7 to 8
1.172	13.60	91.83	40	211	8 to 9
1.170	13.57	91.83	40	212	9 to 10
1.173	14.07	91.83	40	214	10 to 11
1.169	13.68	91.83	40	208	11 to 12 m'dn't
					12 to 1 A.M.
					1 to 2
					2 to 3
					3 to 4

All of the columns in this table will be readily understood. The temperature of the feed-water has been obtained by assuming that the air pump discharge has been reheated to 100° Fahr. in the primary heater, $6\tfrac{3}{4}\%$ of exhaust steam being available in the auxiliary heater, resulting in a temperature of 172° for full-load conditions. Figure 44 has been applied in correcting this temperature for variable load. S_f has been assumed at 100, P at 180, S_c at 40 and E_c at 55%. Wh is figured at 1000. The last column shows results of K_h (equation No. 53, in this case) for the average conditions of each hour. The sum of the figures in this column, or 4195, will equal ΣK_h.

We may now obtain the following:

From Fig. 43 C	....	.97
From Fig. 45 KOL	....	55,000
and KO^2L	....	1,100,000
From Fig. 46 $\dfrac{KO^2L}{C\Sigma K_h}$	....	271.
From Fig. 47 $\dfrac{5.95S}{HE_b}$	....	16.
From Fig. 48 K_{hv}	....	191.

This is the required and final result and equals the kilowatt hours per barrel of oil including all stand-by losses, etc. In case it is desired to apply this method to a long period, say six months, for instance, it would be necessary to reduce the result a small amount which would correspond to the steam wasted in cleaning boilers and in blowing off tubes, etc., which losses are NOT included above.

CONCLUSION

WITH the above developments and examples, this treatise reaches its conclusion. The system has been completely described. But, by way of final remark, it might be stated that notwithstanding the importance of the fuel economy factor in the design of any power station there are other considerations which must be carefully dealt with; local conditions must be met, industrial requirements which demand attention, and other features of cost, reliability, etc., which must be carefully weighed before arriving at a final conclusion. Despite the desirability of a decreased annual fuel bill, of what benefit is such economy if the parts and appliances which go to make it possible also give rise to operating difficulties, require constant and expensive attention and repairs, or render continuous service difficult or impossible?

In other words, a power plant to be a financial and an engineering success, should be capable of continuously and positively caring for its requirements at a minimum total expense. Here is involved the total operating expense of a steam-power plant which is another distinct subject. The items of labor, water, supplies, depreciation, maintenance, interest on the original investment, etc., will all enter into the problem. Thus it develops into a process of apportionment — of assigning relative values to each phase of the problem and finally producing a station, not of absolute necessity the most economical of fuel, but embodying the maximum of each feature tending towards reliability of operation, long life, and small TOTAL upkeep.

Nevertheless, the present cost of fuel — to say nothing of its continual rise — lends so much weight to the problem of fuel economy that this item is easily of the most importance. Provided a steam-power plant containing no complications or intricate devices is employed, and that superior economic results are obtained only by the use of the highest grade of substantial machinery, properly distributed and connected, it is safe to say that in nine cases out of ten, the more economical the installation, the better the investment. This explains the reason for the immense amount of effort spent upon this subject by our leading engineers.

With our final table, this treatise will be concluded. The various formulæ are necessarily scattered to some extent, requiring a certain amount of time and trouble in selecting the particular equation needed.

The notation is also difficult to locate.   Figure 50 sets forth all the principal formulæ developed above, together with the notation employed.
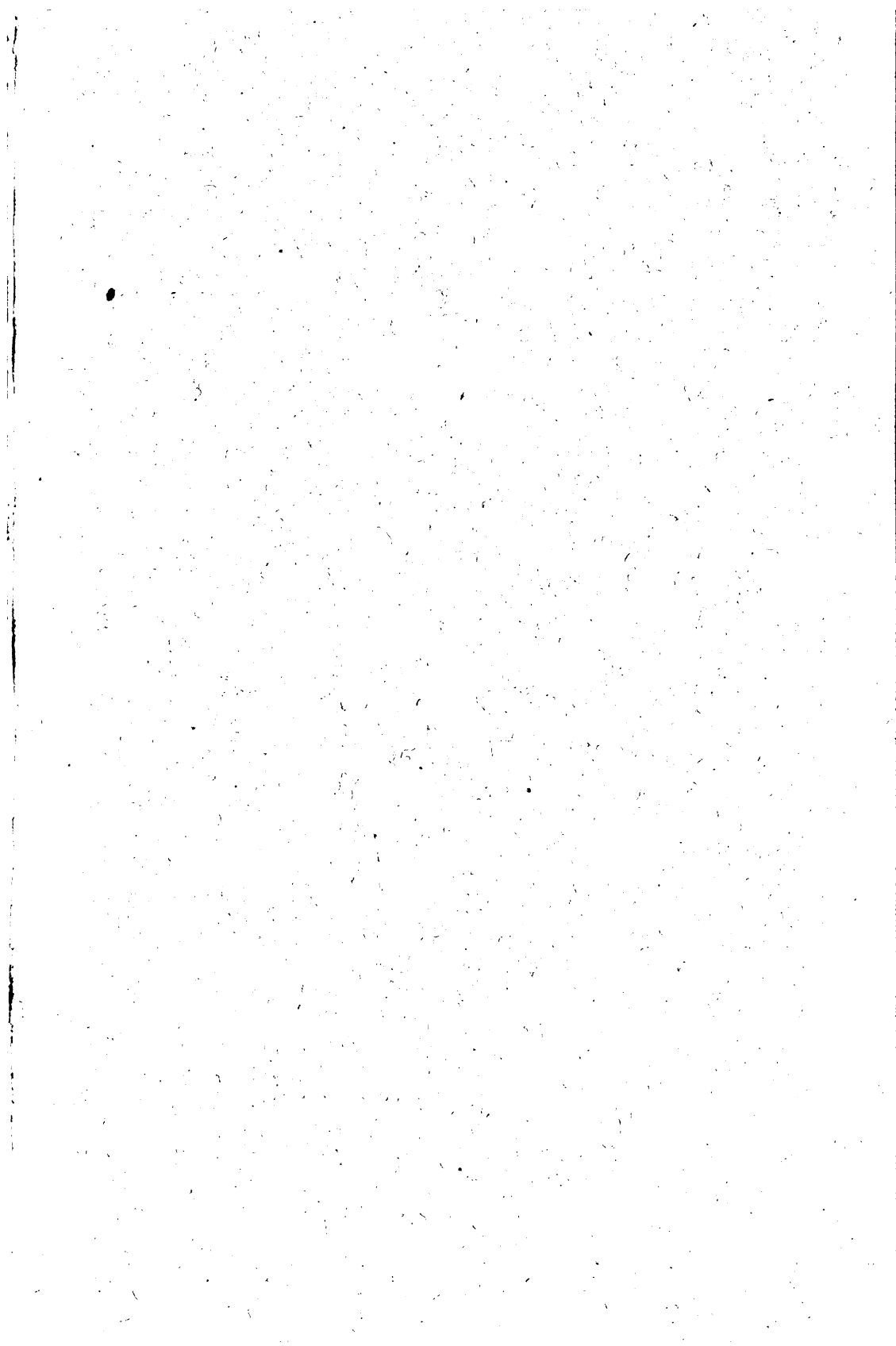
FIG. 50. — RESUMÉ OF FORMULÆ AND NOTATION

Kind of Plant	No. of Equation	Equation
Non-Condensing — P = 150 — S_f = 200 ...	21	$K_h = \dfrac{.23\, HE_bE_eE_g}{FS_e}$
Non-Condensing (any values of P and S_f)..	35	$K_h = \dfrac{HE_bE_eE_g\left(638 - \dfrac{PS_f}{1000}\right)}{2642\, FS_e}$
Surface Condensing (steam-driven auxiliaries)..............	53	$K_h = \dfrac{HE_bE_eE_g\left(638 - \dfrac{PS_f}{1000}\right)}{1.06\, FS_e\left(2500 + S_a + \dfrac{WhS_c}{792\, E_c}\right)}$
Surface Condensing (power-driven auxiliaries)..............	68	$K_h = \dfrac{HE_bE_eE_g\left(638 - \dfrac{PS_f}{1000}\right)}{1.06\, FS_e\left(2500 + \dfrac{S_e}{E_eE_gE_{am}} + \dfrac{WhS_e}{792\, E_eE_gE_cE_{cm}}\right)}$
Jet Condensing (steam-driven auxiliaries)...	80	$K_h = \dfrac{HE_bE_eE_g\left(638 - \dfrac{PS_f}{1000}\right)}{.08\, FS_e\left[33000 + S_a(W+1) + \dfrac{WhS_c}{60\, E_c}\right]}$
Centrifugal or Triplex Pumping Plants (condensing)...........	90	$D = \dfrac{2.51\, HE_bE_eE_p\left(638 - \dfrac{PS_f}{1000}\right)}{FS_e\,(2500 + S_a)}$
High Duty or Direct-Acting Pumping Plants (non-condens'g)	97	$D = \dfrac{D_pHE_b\left(638 - \dfrac{PS_f}{1000}\right)}{1971430\, F}$
High Duty or Direct-Acting Pumping Plants (condensing)..	105	$D = \dfrac{D_pHE_b\left(638 - \dfrac{PS_f}{1000}\right)}{1971.43\, F\,(1000 + S_a)}$
Any Plant (for variable load)...........	112	$K_{hv} = \dfrac{KOL}{\dfrac{KO^2L}{C\Sigma K_h} + \dfrac{5.95\, S}{HE_b}}$

NOTATION EMPLOYED IN DERIVING ABOVE FORMULÆ

K = Rated capacity of plant in Kw.

K_h = Kw.-hours per bbl. oil at full rated load

K_{hm} = Mean value of K_h for the actual number of hours per day of operation

K_{hv} = Kw.-hours per bbl. oil for variable load including " stand-by losses "

E_e = Efficiency of main engine in per cent

E_g = Efficiency of main generator in per cent

E_b = Efficiency of boiler in per cent

E_c = Efficiency of circulating pump in per cent

E_{cm} = Efficiency of circulating pump motor — per cent

E_{am} = Efficiency of air-pump motor in per cent

E_p = Efficiency of main pump in per cent — (for pumping plants)

S_e = Steam consumption of main engine in lbs. per I.H.P.-hour

S_f = Steam consumption of feed-pumps in lbs. per I.H.P.-hour

S_a = Steam consumption of air-pumps in lbs. per I.H.P-hour

S_c = Steam consumption of circulating pumps in lbs. per I.H.P.-hour

p = Total steam per hour required by feed-pumps in lbs.
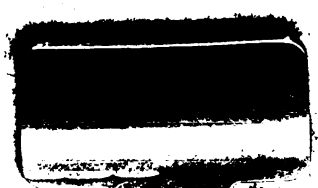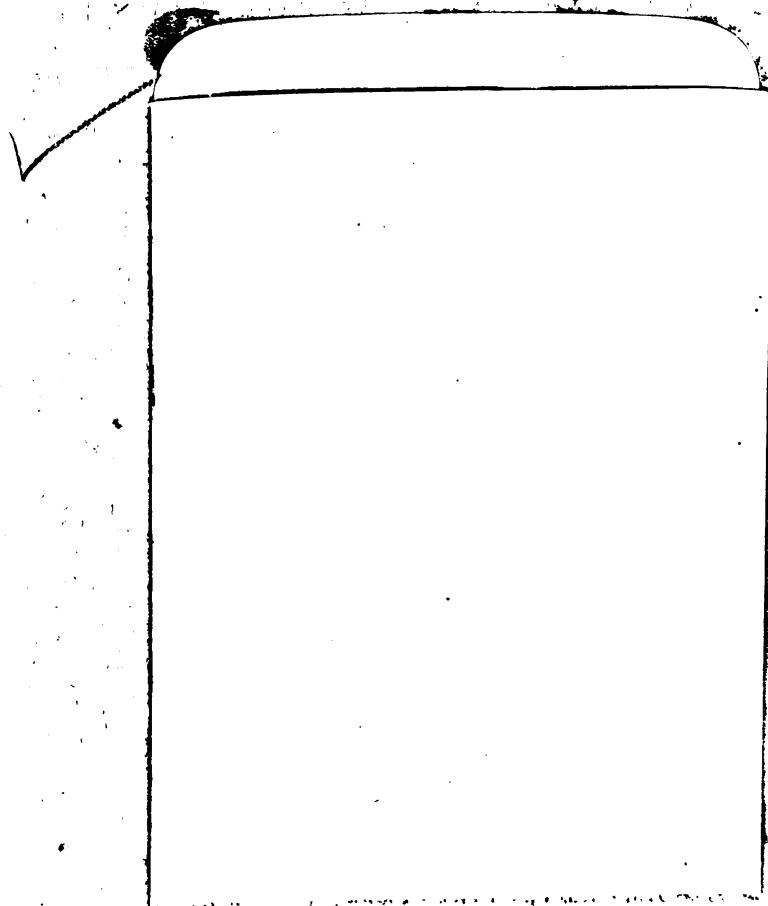
P = Boiler pressure by gage in lbs.

x = Total steam per hour required by air-pumps in lbs.

y = Total steam per hour required by circulating pumps in lbs.

H = Calorific value of fuel oil in B.T.U. per lb.

F = Factor of evaporation

W = Lbs. of circulating or injection water per lb. steam

h = Total head on circulating or injection pump in feet

G = Total U. S. gallons of water pumped per minute (pumping plants)

T = Total head on main pump in feet (pumping plants)

D = Duty of complete plant expressed in millions of foot-pounds of work performed per bbl. oil burned

D_p = Duty of main pumping engine — millions of foot-pounds per 1000 lbs. steam sup'd to main eng.

O = Hours of operation per day

L = " Curve load factor " in per cent

C = Radiation and leakage coefficient

S = " Stand-by " loss expressed in total boiler H.P.-hours idle per day, but under banked fires

Σ = The process of summation

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